THE GAS TURBINE.

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THEORY, CONSTRUCTION, AND RECORDS OF THE RESULTS OBTAINED FROM TWO ACTUAL MACHINES.

BY

HANS HOLZWARTH, Engineer.

TRANSLATED BY

A. P. CHALKLEY, B.Sc. (Lond.), A.M.Inst.C.E., A.I.E.E.,

WITH ADDITIONAL NOTES BY THE AUTHOR,

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PREFACE.

THE gas turbine described in this book is the result of the collaboration of the author, as technical designer, and Mr Erhard Junghans of Schramberg, who undertook the cost of the work.

After the necessary steps had been taken to ensure the patent rights in Germany and abroad, it was no longer necessary to delay publication of the reports on this gas turbine.

Information concerning the systematic investigation work, which has now been proceeding for more than three years, on an important technical problem should not fail to be of great interest, both theoretically and practically, to a large circle of engineers.

The author desires at this point to express his appreciation of the help given by Mr Erhard Junghans of Schramberg, who has directed the business part of the undertaking. Mr Erhard Junghans has borne the entire and substantial financial burden, keeping the undertaking on an independent basis, and never once expressing his doubt regarding its success.

The extent to which this business control aided in, and advanced, the development of the gas turbine will be evident to all who have been engaged on such work of development.

The original turbine was built to my designs by Messrs Körting Bros. of Hanover; the first turbine employed for actual driving, also after my designs, was constructed by Messrs Brown, Boveri & Co. of Baden-Mannheim, who also supplied the dynamo and blowing plant.

Both firms allowed me to carry out the experiments uninterruptedly in their famous and well-equipped works.

Messrs Brown, Boveri & Co. placed their wide experience in the construction of steam turbines at my disposal, in the construction and tests of the large plant.

Messrs Julius Pintsch, Berlin, supplied the gas plant, and Messrs Robert Bosch, Stuttgart, the combustion apparatus for the large test plant at Mannheim. These firms helped by their combined efforts to bring the gas turbine to the point of success as a purely German invention.

Mr Fritz Pfaller helped me very considerably both with the drawings and in carrying out the experiments.

HANS HOLZWARTH.

MANNHEIM,
10*th November* 1911.

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THE GAS TURBINE.

PART I.

THEORY OF THE GAS TURBINE.

1. GENERAL DESCRIPTION OF THE GAS TURBINE PROCESS.

As is shown in the tests described later, the following conditions are essential for a practicable gas turbine :—

1. Operation with periodical combustion (explosion), as with reciprocating gas engine.

2. The space in which combustion takes place (combustion chamber) must be closed on the exhaust side leading to the actual turbine, at least during the greater portion of the charging with the combustible mixture.

3. The combustion chamber must be closed on the admission or charging side during the combustion and expansion period.

4. The combustion chamber must be scavenged with air during the interval between the individual explosions.

For a gas turbine complying with each of these conditions, and provided with a combustion chamber, it is equally satisfactory for the closing arrangements (nozzles, valves, or flaps) to be on the exhaust or delivery side, or for the admission or charging side to be provided with mechanically operated valves which serve to deliver air into the combustion chamber.

Referring to fig. 1 :—

A is the combustion chamber.
B is the air reservoir or chamber.
C is the gas reservoir or chamber.

D is the mechanically operated air inlet valve.
E is the mechanically operated gas inlet valve.
F is the nozzle valve (flap valve).
G is the nozzle.
H is the rotor.
J is the exhaust.

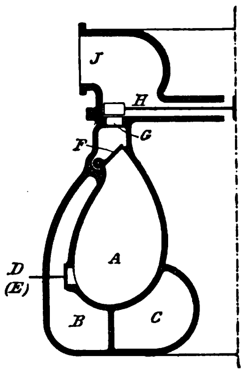


Fig. 1.—Diagrammatic sketch of the gas turbine.

Several such combustion chambers are arranged in a circle. They are charged successively at predetermined intervals.

The chambers B and C are kept filled with air and gas respectively at low pressure by means of a compressor of any type, driven in a convenient manner, and a vacuum is maintained at J by means of an exhauster. Through the inlet valves D and E, air, and afterwards gas, are admitted into the combustion chamber. When A is filled with air, gas is blown in, and, owing to the eddying thus produced, the air gas becomes thoroughly and intimately mixed. During this operation the nozzle valve F remains closed. Immediately after the explosion the pressure thereby produced causes F to open, and the gases attain a velocity, in passing through the fixed nozzle G, corresponding to the velocity energy abstracted from the available energy contained in the exploded gases.

This velocity energy is used in the rotor H, and the spent gases are exhausted through J.

After the passage of the expanded gases, the nozzle valve F is closed mechanically, slowly enough to allow the scavenging air entering through D to clear effectively the chamber A, and provide sufficient cooling air to the turbine.

2. GENERAL ANALYSIS OF THE IGNITION, EXPLOSION, COMBUSTION, AND EXPANSION PROCESSES.

The theory of the gas turbine will be developed as it was proved step by step by the experiments.

Speaking generally, the gas and air mixture passes through the following cycle in the gas turbine:—

Immediately before ignition the mixture in the combustion chamber is in the condition $p_0 v_0 T_0$.

The mixture is ignited at various points simultaneously, and combustion takes place, and its condition is then represented by $p_1 v_1 T_1$.

It is assumed that, if this takes place regularly and evenly (not suddenly by explosion), the total available heat is set free during combustion. Up to this time all the changes occur at constant volume.

When the condition $p_1 v_1 T_1$ is reached, the connection to the turbine is suddenly opened.

The high-pressure gases are then expanded in the nozzle to the condition $p_2 v_2 T_2$, and their available energy is converted into velocity energy; and in the turbine itself this velocity energy is converted into mechanical work.

When the condition $p_2 v_2 T_2$ is reached, the gases are no longer concerned in the working cycle, which is completed on the exhaust side of the rotor.

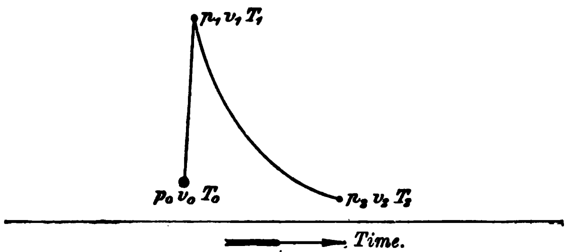


FIG. 2.—Ignition, combustion, and expansion diagram.

These changes of condition will be examined in a general analytical form, without going into too much detail, mainly by means of the entropy diagram.

Previous to the changes of state as described above, the combustion chamber is scavenged and cooled with the gas and air mixture.

The process of charging and scavenging, and the working cycle in the turbine itself, will be examined in their general theoretical aspect in the next section.

In all these changes of state the fundamental formulæ for specific heat are employed:

$$c_v = a_v + bT$$
$$c_p = a_p + bT.$$

Conditions immediately before Ignition.—The chamber is filled with—

1. g kg. of gas of which the heat content is H, of specific gravity γ_1 at 0° C. and 1 atmosphere pressure (abs.); and

2. l kg. of air of specific gravity γ_2 at 0° C. and 1 atmosphere pressure (abs.).

If the capacity of the combustion chamber is V cubic metres, the volume of the gas is

$$v_1 = \frac{g}{\gamma_1} \times \frac{p_1}{p_0} \times \frac{T}{273},$$

and the volume of the air for combustion, in kg., is

$$v_2 = V - v_1,$$

and the weight l of the air for combustion, in kg., is

$$l = v_2 \gamma_2 \frac{p_0}{1} \times \frac{273}{T_0}$$

$$= V \gamma_2 p_0 \frac{273}{T_0} - g \frac{\gamma_2}{\gamma_1}.$$

The weight of the charge is

$$G = g + l.$$

The ratio of the weight of gas to weight of air in the mixture is

$$1 : \frac{l}{g},$$

and the available quantity of heat per kg. of the mixture is

$$Q = \frac{gH}{g+l}.$$

Combustion.—The heat set free may be expressed as follows:

$$Q + \int_{273}^{T_0} c_v' dT = \int_{273}^{T_1} c_v dT + Q_w',$$

in which c_v' is the specific heat of the mixture before the combustion, and Q_w' the quantity of heat lost during combustion by radiation and conduction.

The above formula can also be written:

$$\int_{273}^{T_1} (a_v + bT) dT = Q + \int_{273}^{T_0} (a_v' + b'T) dT - Q_w'$$

$$= Q_1 - Q_w',$$

$$\frac{b}{2} T_1^2 + a_v T_1 - (Q_1 - Q_w' + 273 a_v + \frac{b}{2} 273^2) = 0,$$

$$T_1 = \frac{-a_v + \sqrt{a_v^2 + 2b(Q_1 - Q_w' + 273 a_v) + (273 b)^2}}{b}.$$

The highest pressure during combustion (explosion) is

$$p_1 = \phi p_0 \frac{T_1}{T_0}.$$

ϕ = coefficient of contraction of the chemical action.

Expansion.—Let Q_a be the quantity of heat which is rendered available by the expansion, to do useful work in the turbine; Q_e, the heat remaining in the exhaust gases after the expansion; and Q_w'', the heat which is carried away by the walls through radiation and conduction during expansion; then

$$Q_a = Q_1 - (Q_w' + Q_w'' + Q_e)$$

$$= \int_{T_2}^{T_1} (a_v + bT)dT - Q_w'',$$

and

$$Q_w'' = \int_{S_2}^{S_1} TdS,$$

in which S_1 and S_2 are the entropy values at points (1) and (2) respectively, and

$$Q_e = \int_{273}^{T_2} (a_v + bT)dT.$$

The transmission of heat to the walls of the combustion chamber is given by the general equation for the transmission of heat by radiation: *

$$Q_r = CFz\left[\left(\frac{E}{100}\right)^4 - \left(\frac{T}{100}\right)^4\right],$$

in which C = 4 for unpolished cast-iron,

z = time in hours,

F = surface in square metres,

E = absolute temperature of heat-absorbing parts,

T = absolute temperature of heat-radiating parts.

For the loss through conduction,

$$Q_c = C_1 Fz(E - T),$$

in which the values are as above, except the constant.

For gas in a state of rest,

$$C_1 = 4.$$

For gas in motion with a velocity of v^m along the wall,

$$C = 2 + 10\sqrt{v^m}.$$

The total heat transmitted by the walls is

$$Q_w = Q_r + Q_c.$$

The efficiency of combustion, that is, the ratio of heat rendered available for useful work by combustion, to the heat actually given to the cycle is

$$\eta = \frac{Q_a}{Q}.$$

The general determination of T_2 by purely analytical methods is rather complicated. In the fifth edition of Schlöttler's book on *Gas-*

* Stefan-Boltzmann's equation.

maschine, on page 382, he develops the formula for T_2 for the special case:

$$Q_w'' = \int_{S_2}^{S_1} T dS = 0$$

for pure adiabatic expansion

The formulæ are then:

$$\text{if } \epsilon = \frac{v_2}{v_1} \text{ (compression ratio)}$$

$$\text{and } K' = \frac{a_v}{a_p},$$

$$\log \epsilon = \left[\log \frac{p_1}{p_2} + \frac{bT_1}{a \log \text{nat. } 10}\left(1 - \epsilon\frac{p_2}{p_1}\right)\right] : K'$$

$$T_2 = \epsilon T_1 \frac{p_2}{p_1}.$$

Stodola, in his paper (*Z. d. V. d. I.*, 1898, p. 1045 *et seq.*), explains clearly and concisely the combustion cycle in a general form, by means of the entropy diagram.

Since the exchanges between the mixture and the walls are of great importance for the gas turbine process, and these exchanges can be examined more clearly in entropy diagrams than by a purely analytical method, it will suffice in the analysis if the theoretically deduced results are in general accordance with those obtained experimentally, without attempting to bring all the simpler formulæ with their distinct limitations into unison.

3. ENTROPY DIAGRAMS FOR IGNITION, COMBUSTION, AND EXPANSION.

Entropy Diagram.

The entropy diagram in the form given by Stodola may be employed in the following manner to examine the process in the gas turbine.

The co-ordinates for equal pressure and equal volume are drawn for the kilogram-molecule, accepting the values of Mallard and Lechatelier, which are now generally agreed to be rather too high.

Carbon dioxide	$= 4{\cdot}39 + 0{\cdot}00774T$
Steam	$= 4{\cdot}21 + 0{\cdot}00574T$
Permanent gas	$= 4{\cdot}10 + 0{\cdot}00244T.$

The state of the gas represented by $p_0 v_0 T_0$ corresponds to the point A in fig. 3; this is the starting-point of the cycle.

From the point O derived from the entropy constant the straight lines (lines of direction) are drawn for the specific heat:

OD for $c_v = d_v$ (for a perfect gas),
EF parallel to $c_v = a_v$

(a_v is to be reckoned from the molecular combining weight of the air and

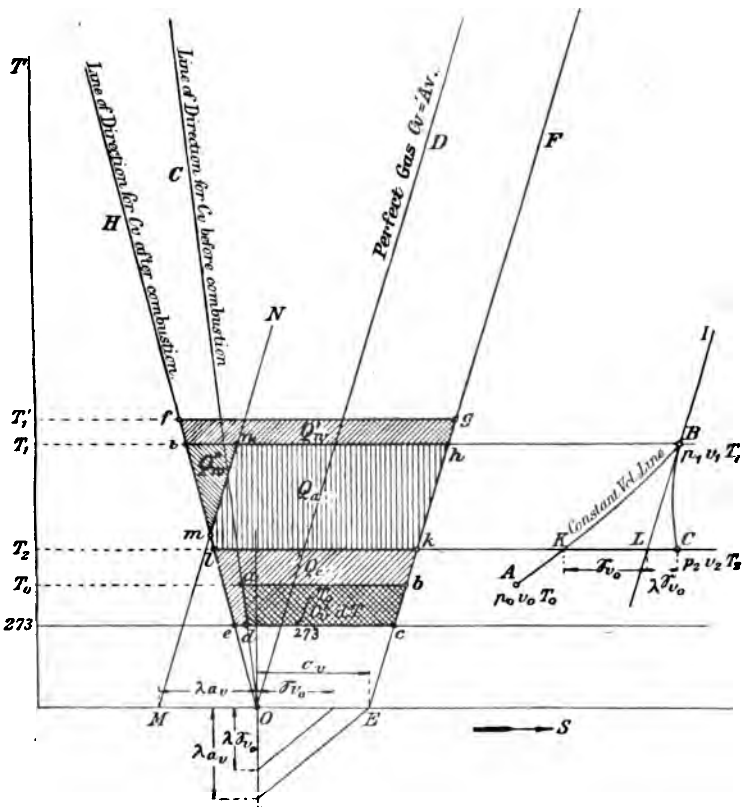


FIG. 3.—Sketch showing entropy diagram.

gas mixture; in Stodola's diagram $a_v = 4{\cdot}15$, which is taken as constant for all gases),

OG for $c_v' = a_v + b'\mathrm{T}$

for the gas before combustion.

The variables b' and b are determinable from the molecular combining weights by known methods.

The trapezoid $a\,b\,c\,d\,a$ enclosed between the lines $\mathrm{T} = 273$ and $\mathrm{T} = \mathrm{T}_0$ and the direction lines OG and EF represents the quantity of heat

$$\int_{273}^{\mathrm{T}_0} c_v' d\mathrm{T},$$

that is, the quantity of heat in the mixture at the moment before the heat Q is set free.

This heat Q is set free by ignition, and is represented by the area

$$a\,d\,e\,f\,g\,b\,a = Q.$$

If the scale of areas for the quantity of heat per kg. or kg.-molecule is known, the line $f\,g$ is deduced, and therefore the top temperature T_1'.

The trapezium $e\,f\,g\,c\,e$ gives the quantity of heat

$$Q_1 = Q + \int_{273}^{T_0} c_v' dT,$$

i.e. the total quantity of heat in the mixture after ignition.

During the explosion (a period of about $\frac{3}{100}$ of a second in practice) some of the heat passes to the walls. The quantity of this heat Q_w' can be averaged from the formulæ:

$$Q_w' = Q_r + Q_c$$

$$Q_r = CFz\left[\left(\frac{E}{100}\right)^4 - \left(\frac{T}{100}\right)^4\right]$$

$$Q_c = C_1 Fz[E - T].$$

In these formulæ the following values may be substituted:

$C = 4.$
F = internal surface of combustion chamber in square metres.
$z = \frac{1}{120{,}000}$, about.
E = the temporarily constant gas temperature.
T = the temperature of the walls.
$C_1 = 4$ (the gas being in a state of rest).

Calculation in individual cases shows that the heat Q_c' carried away by conduction is negligible compared with Q_r', as this value depends on the 4th power of the temperature of the gas; the total Q_w' is in most cases small compared with Q_w'', because of the short period of time.

Treating the general case, Stodola subtracts the value

$$Q_w' = \text{trapezium } f\,g\,h\,i\,f$$

from

$$Q_1 = f\,g\,c\,e\,f.$$

In this way the value T_1 when the state of the gas is $p_1 v_1 T_1$ is obtained. The intersection of the line T_1 with the volume curve through $p_0 v_0 T_0$ gives the point $p_1 v_1 T_1$.

Up to this stage the purely analytical analysis is not particularly complicated.

Strictly speaking, Q_w' should not simply be subtracted from the top of the trapezium, but a narrow wedge-shaped piece should be deducted, with one side along the line *f e i*, and increasing rapidly in width in the upward direction.

Omitting Q_c', and remembering that E^4 is unimportant in relation to the 4th power of the temperature of the walls, the following is approximately true:

$$dQ_w' = C \times T^4 \times dz.$$

The extent to which the temperature is dependent on the time of combustion is not exactly known (*vide* Schlöttler, p. 464). Assuming that T varies directly with z, or

$$T = az,$$

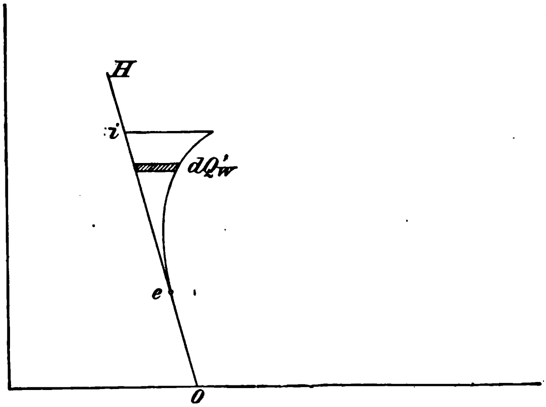


FIG. 4.—Sketch showing Q_w'.

the above formula may be written

$$dQ_w' = C \times T^4 \times dT$$

and

$$Q_w' = C(T_1^5 - T_0^5).$$

T_0^5 is negligible compared with T_1^5, and therefore it is correct to subtract the total value Q_w' as the area *f g h i f* in the entropy diagram from the top of the trapezium Q_1.

The trapezium *e i h c e* therefore represents the total heat present in the gases after complete combustion.

Stodola obtains the relation between the heat carried away by the walls during the polytropic expansion with the expansion line BC of the entropy diagram by dividing the line CK into two parts with a parallel to AD, so that

$$KL = X_{v0}$$
$$LC = \lambda X_{v0},$$

and further making

$$(OM) = \lambda(OE)$$

and drawing a parallel to OD through M. This parallel cuts off the area *i n m i* from Q, and this represents the heat Q_w'' which determines the variation from adiabatic expansion; in the entropy diagram adiabatic expansion would be represented by a parallel to OH.

The point C is not known, except that it must lie on the line p_2.

In practice, the simplest way to proceed is to take C and then draw MN, and calculate the area *i n m i*. Or

$$Q_w'' = Q_r'' + Q_c''$$

is calculated from the formulæ, referred to the expansion period.

This factor is dependent only on the nozzle opening; the expansion period is, of course, made as small as is allowable, in order to reduce Q_w'' as far as possible. This point naturally influences to a large extent the design of the combustion chamber and the nozzle valves.

In practice, the most satisfactory time for expansion is 0·2 to 0·4 second.

Since, when C is assumed, T is known, it follows that Q_r'' can be calculated as

$$E = \sqrt{T_1 T_2}\,;$$

the temperature of the walls is unimportant in this case.

The value of Q_c'' is not easily determined. The portion which is carried away in the actual combustion chamber walls through conduction is comparatively small, as the gases can be in a state of rest. The coefficient of conduction on the way to the nozzles is much larger, as the gases are in motion; but, on the other hand, the heat-receiving surface is small.

Representing the calculated quantity of heat,

$$Q_w'' = Q_r'' + Q_c'',$$

by the area *i n m i*, the process being continued until the heat is constant, then C represents the conditions at the end of the expansion, and the area *m n h k l m* represents the quantity of heat available for the working cycle in the turbine.

The trapezium *l k c e l* shows the amount of heat, Q_e, and the fraction $\dfrac{Q_a}{Q}$ gives the *efficiency of combustion,* which is the same as the indicated efficiency of the reciprocating engine, if the questions of delivering the gas mixture in the state $p_0 v_0 T_0$, and discharging it in the atmosphere in the state $p_2 v_2 T_2$, be neglected.

From the foregoing the importance of any deviation from the adiabatic expansion line is apparent.

It is a mistake to assume that the heat given to the walls necessarily reduces Q_a, and diminishes the efficiency. If heat passes to the walls, T_2 is reduced and the line KL is lowered, so that

$$\int_{T}^{T_1} c_v dT = \text{trapezium } i\,h\,k\,l\,i$$

increases in relation to

$$\int_{T_2}^{T_1} c_v dT$$

with adiabatic expansion.

The passage of heat to the walls therefore takes place, not only at the expense of Q_a, but at the combined expense of

$$Q_w'' = Q_a'' + Q_e''.$$

The heat Q_w'' must, of course, be deducted from the increased area

$$\int_{T_2}^{T_1} c_v dT.$$

The above general remarks have been made owing to the funda-

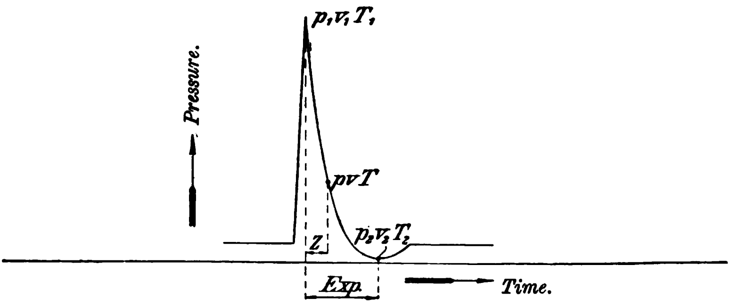


FIG. 5.—Pressure-time diagram.

mental importance of the passage of heat to the walls in the gas turbine. It may therefore be stated how this influence may be more rigidly investigated.

On a pressure-time diagram taken from the combustion chamber of a gas turbine, let the pressure fall after z seconds (reckoned from the state $p_1 v_1 T_1$). The expansion may be divided into sections; if, for instance, the total duration of the expansion is 0·3 second, the various states might be:

after 0·1 sec.,		$p_2' v_2' T_2'$,
„	0·2 „	$p_2'' v_2'' T_2''$,
„	0·3 „	$p_2 v_2 T_2$.

The point C corresponding to $p_2' v_2' T_2'$ is next obtained as above (fig. 6). Then, by Stodola's method, the trapezium $i\, n' m' o'$ will correspond to the amount of heat given to the walls during the partial expansion, and similarly for C″.

As the transference of heat through radiation is again of great importance, the trapezium $i\, o' m' n' i$ does not completely represent Q_w'', but the line $n'm'$ must be replaced by a curve, which represents at least the 5th power of T. The area, however, remains unaltered.

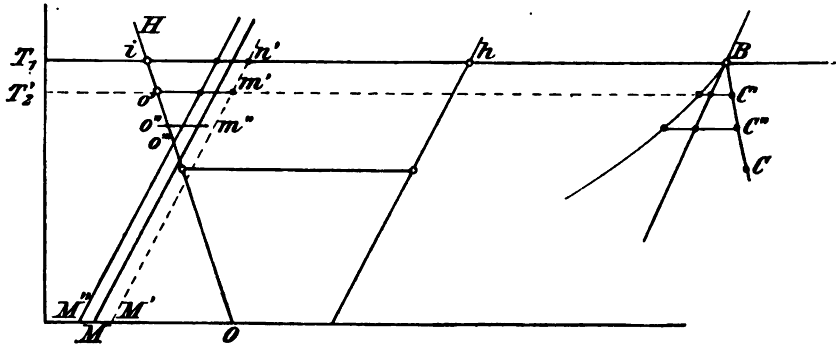


FIG. 6.—Entropy diagram sketch.

The process may be represented on the entropy diagram as under.

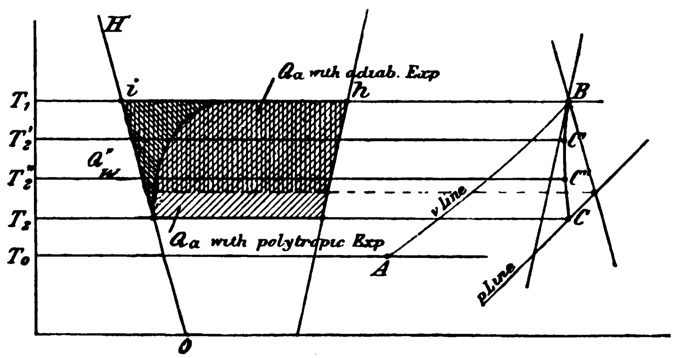


FIG. 7.—Sketch referring to entropy diagram.

The vertically hatched area represents Q_a with adiabatic expansion.

4. WORKING CYCLE IN THE ACTUAL TURBINE.

Having determined Q_a and η, the next point of interest is the method of utilising this available heat in the actual turbine. This turbine has one pressure stage and one to three velocity stages, according to circumstances. More pressure stages are impracticable, first, because the scavenging air would have a much lower pressure drop through the nozzles, and, secondly, because the opportunity for heat radiation from the walls would become too great.

The type and method of admission to the turbine is the same as for all intermittent-action steam turbines, and, in particular, the pure Parsons turbine.

This works with blasts of steam, *i.e.* at regular intervals the space between the inlet valve and the first ring of guide blades is filled with live steam, after which the inlet valve is closed or throttled according

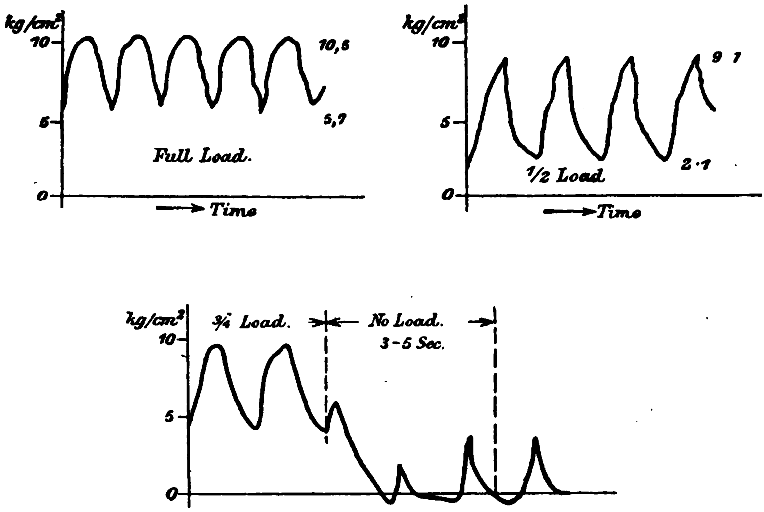


FIG. 8.—Indicator diagrams of Parsons turbine.

to the load, and the steam expanded in its passage through the turbine. In the combination type of Parsons turbine, in which the high-pressure portion is designed as a pure Curtis turbine with nozzles and multi-stage impulse wheels, the principle of steam blast admission has been retained; the blasts are, however, less pronounced than in the pure Parsons type (fig. 8*a*).

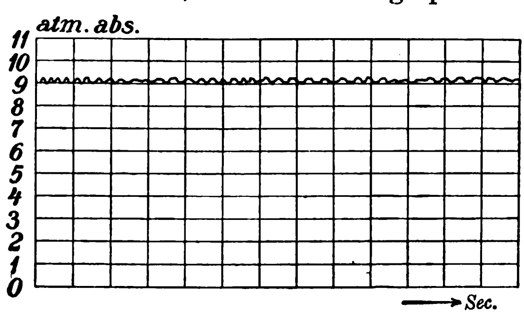


FIG. 8*a*.—Modern Parsons turbine.

The gas turbine works in a similar manner. At the moment of complete combustion, the chamber is filled with the gas whose kinetic energy is Q_a. As the quantity of heat decreases, the gas passes out with a diminishing velocity until the heat drop is complete, and there is a fresh blast after scavenging and charging.

Consider the process with intermittent expansion (c_v) instead of continuous (c_p), and let

$$\frac{Q_a'}{A} = H,$$

the available drop in metre-kilograms at a particular moment, and

C = the velocity of flow at this moment,
ϕ_n = the nozzle efficiency,
ϕ_b = the blade efficiency,
ϕ_e = the exhaust efficiency;

then $\eta_t = \phi_n \phi_b \phi_e$ = turbine efficiency.

The air resistance can be included in the blade losses. Then

$$\int_0^{G=1} AHdG = \int_0^{G=1} Q_a' dG = Q_a,$$

which is the work actually performed during the explosion by G kg. of gas:

$$L = \int_0^G \eta_t H dG$$

and

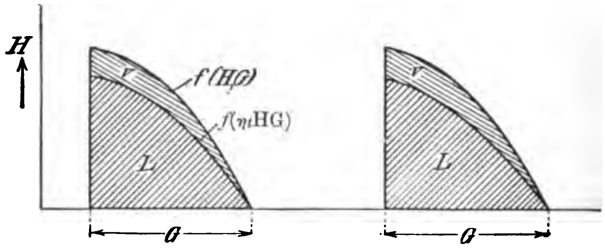
$$dG + F\gamma_2 c dz,$$

in which F is the nozzle opening at exit, and γ_2 the specific heat of the gas in the condition $p_2 v_2 T_2$, and

$$\phi_n \frac{c^2}{2g} = H.$$

If the G – H diagram is drawn, we obtain the following, with intermittent working:

$$L = \int_{T_2}^{T_1} c_v dT - Q_w''.$$



Total weight of charge.

FIG. 9.—$L = \int HdG$ curve; c_v.

The area L represents the actual work performed by a charge of G kg., and the area V represents the nozzle, blade, and exit losses; while the area L + V represents the work performed by a charge of G kg. in a perfect engine.

If G = 1, then L + V = Q_a.

With continuous operation, when

$$L = \int_{T_2}^{T_1} c_p dT - Q_w'',$$

the diagram is considerably altered.

The time which the charge of G kg. takes for combustion comes into consideration in the calculation of the horse-power. It is, therefore, not correct to compare the work performed in m.kg. in intermittent operation with a machine in which the whole charge of G kg. passes out with the maximum drop H. To do so involves a similar fallacy to confusing c_p with c_v. This would mean that a capacity for work would be ascribed to the nearly perfect engine which it by no means possesses. Further,

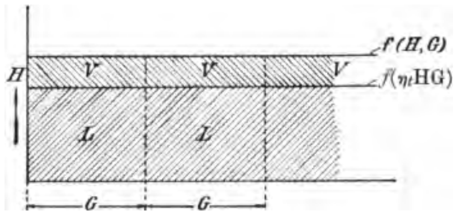


FIG. 10.—L curve; c_p.

$$\eta = \frac{L}{L+V} = \frac{\int_0^G \eta H dG}{\int_0^G H dG}$$

η_t

and

$$\eta \text{ total} = \eta \eta_k,$$

where η_k = the efficiency of combustion.

The type of the curve f(HG) depends on that of the expansion curve $a\,b$.

A calculation and representation of the total expansion, divided into partial expansions, may be made of value by taking actual figures.

The question of the dimensions of the nozzles depends to a great extent on the time during which the exploded mixture of gas and air expands through the nozzles. As will be shown later, it is preferable, or in fact necessary, to make this time as short as possible in order to minimise the loss due to radiation of heat. For this reason it is advantageous to make the smallest opening of the valve less than $\frac{5}{1000}$ of the capacity of the combustion chamber divided by the diameter of the sphere of equal capacity.

As regards the form of the nozzle itself, and particularly the ratio of the section at the outlet to the section of the most constricted part, it may be mentioned that it is desirable for the total velocity to be developed in the nozzle itself.

If the mixture leaves the nozzle with a velocity which is not the maximum attainable, this maximum is reached either in the clearance space or in the first moving wheel, so that there will be small drop of pressure. In steam turbines this is not of great importance; but the contrary is the case with gas turbines, in which the scavenging air should pass through the nozzle and around the blade passages with as little obstruction as possible.

It is therefore of importance, particularly during the first portion of the expansion where there should be a large heat drop, that the maximum velocity should be reached in the nozzles. The proportions of the characteristic nozzle sections have to be fixed correspondingly. The formulæ used are as under:—

Let c_m' = the momentary velocity in the smallest section of the nozzle,
T_m' = the corresponding absolute temperature,
c' = the velocity of flow at the exit of the nozzle,
F_m and F = the corresponding sections,
v_m' and v_2' = the corresponding specific volumes,
p_m', p_1', and p_2' = the corresponding pressures,
T_1' and T_2' = the initial and final absolute temperatures.

$$c_m' = \sqrt{2g\frac{K}{K+1}p_1'v_1'}$$
$$= \sqrt{2g\frac{K}{K+1}RT_1'}$$
$$c_m' = 18{\cdot}45\sqrt{T_1'} \text{ approximately,}$$

since $K = 1{\cdot}37$ and $R = 30$.

Also $F_m c_m' = \Delta G v_m'$

and $Fc' = \Delta G v_2'$;

therefore $$\frac{c_m'}{c'} = \frac{F}{F_m} \times \frac{v_m'}{v_2'} = \frac{F}{F_m} \times \frac{T_m'}{P_m'} \times \frac{p_2}{T_2'}.$$

Now $$p_m' = \left(\frac{2}{K+1}\right)^{\frac{K}{K-1}} p_1'$$
$$= 0{\cdot}523 p_1' \text{ approximately}$$

therefore $$\frac{c_m'}{c'} = \frac{F}{F_m} \times \frac{T_m'}{{\cdot}523 p_1'} \times \frac{p_2}{T_2'},$$

or $$\frac{F}{F_m} = \frac{c_m'}{c'} \times \frac{{\cdot}523 p_1'}{T_m'} \times \frac{T_2'}{p_2}.$$

Further $\frac{T_1'}{T_m'} = \left(\frac{p_1'}{p_m'}\right)^{\frac{K-1}{K}} = \left(\frac{1}{\cdot 523}\right)^{\cdot 27}$

$= 1{\cdot}19;$

therefore $T_m' = {\cdot}84T_1',$

and $\frac{F}{F_m} = \frac{c_m'}{c'} \times \frac{\cdot 523p_1'}{\cdot 84T_1'} \times \frac{T_2'}{p_2}$

$$= {\cdot}624\frac{c_m'}{c'} \times \frac{p_1'}{p_2} \times \frac{T_2'}{T_1'}$$

$$= \text{Constant} \times \frac{T_2'}{\sqrt{T_1'(T_1' - T_2')}} \times \frac{p_1}{p_2}.$$

This formula is contradictory to the assertion that the nozzle efficiency of an explosion gas turbine must be poor, since the ratio of pressures $\frac{p_1'}{p_2}$ steadily decreases, while the ratio of the nozzle sections remains constant. The assertion is based on the generally accepted rule of steam condition that the proportions of the nozzle sections vary directly with the pressure ratio (*vide* B. Loschge, *Z. f. d. g. T.*, 1911, p. 244).

The product

$$\frac{T_2'}{\sqrt{T_1'(T_1' - T_2')}} \times \frac{p_1'}{p_2}*$$

is important in relation to the section ratio in the nozzles. Only p_2 remains constant during expansion, and all the other factors decrease with increasing expansion.

It is easy to determine in individual cases, by means of the entropy diagram, to what extent this characteristic expression for the nozzle ratio varies.

5. SCAVENGING.

After the expansion of the burnt gases to the condition $p_2v_2T_2$ is complete, the condition in the combustion chamber is also $p_2v_2T_2$. For the working process it is immaterial whether the residual gases, which are in the same state as that obtaining behind the nozzles, are in the chamber or the turbine. That is already taken account of in the working cycle; the drop becomes continually less in the last portions of the charge.

These residual gases must, however, be expelled through the nozzle, which requires some energy, and scavenging is accomplished with air. This air can be blown in with a scavenge pump, or drawn in by means

* T_2' is employed because of the heat radiation during expansion.

of an exhauster, and in each case the scavenge air and exhaust gases are discharged into the atmosphere.

The air valve is opened immediately after expansion is complete, and the nozzle valve is still held open; the residual gases in the chamber are then under the influence of the difference of pressure between the scavenge air in front of the scavenge valve ($p_s v_s T_s$) and the conditions prevailing behind the nozzle ($p_2 v_2 T_2$).

At the moment the air valve opens, the state of the residual gases is represented by $p_2 v_2 T_2$, which is altered to pvT by the effect of the scavenge air entering the air valve; the pressure in the chamber is increased, and the temperature lowered, both of which tend to reduce the volume v. The drop is therefore increased, and the residual gases are expelled through the nozzle, containing a larger proportion of scavenge air the longer the scavenging process continues, until finally only pure air flows through the nozzle. The scavenging is continued until the chamber is filled with pure air (which serves as the air for combustion in the next cycle), and the temperature of the mixed scavenge air and exhaust gases corresponds to the working temperature of the turbine, namely, 300° to 400° C.

It is advantageous that the air valve should be opened immediately the state $p_2 v_2 T_2$ is reached, in which case the entrance of the air aids the flow through the nozzle then taking place due to expansion.

An ordered sequential operation of the chambers is also favourable to efficient scavenging; for instance, the flow of gas from one chamber should occur between the passage of the cooling and scavenging air from adjacent chambers on the left and right. This arrangement corresponds to the scavenging action used by the English engine manufacturers. The scavenging air naturally abstracts heat from the walls of the combustion chamber to a greater or smaller extent, according as the temperature of the walls is higher or lower.

These actions are not easy to analyse except by making certain assumptions, as, for instance, that the walls receive no heat, that no mixing with the exhaust gases occurs, that the scavenge air acts as a piston on the exhaust gases, etc.; but with these assumptions the practical results would not be in accordance. The actions are not simple, but attention may be paid to two points which are of great importance to the gas turbine.

The residual gases and scavenge air have not only to pass through the nozzle, but also the guide and moving wheels, whose blading is, however, arranged for the higher velocity of the live gas. The exit velocity of the exhaust gas is, of course, small in proportion, and not sufficient for it and the scavenge air to pass through the blade passages by their own energy. They would be impeded at the inlet edges of the blades and be forced outwards. The practical point following upon this, is that peripheral passages must be arranged through which the exhaust gas and scavenge air can flow without having to go through the blade passages. This is not a disadvantage for the working gas with pure velocity stages, since its own velocity is sufficient to prevent any obstruction in front of the inlet edges of the blades, and the gas passes through the blade passages in the natural manner.

The other point is in reference to the extraction of the heat by the scavenge air from the walls of the combustion chamber. As is clearly shown later in the calculations for special cases, it is of the utmost importance that the temperature before ignition should be as low as possible consistent with proper combustion of the gas, *i.e.* so long as the vaporisation of liquid fuel can take place.

The temperature T_0 may be kept low by having the temperature T_s of the scavenging air low, also by reducing the flow of heat from the walls to the cooling air, that is, by keeping the temperature of the walls low. Whilst, therefore, this temperature of the walls has practically no influence on the cycle during the combustion and expansion phases, and is almost without effect on the amount of heat Q_w' and Q_w'' abstracted by the walls, it has a fundamental influence on the condition $p_0 v_0 T_0$, and therefore on the condition immediately prior to ignition.

The higher T_0 becomes, the greater are T_1, Q_w', and Q_w'', and the effect increases as the 5th power of T_0. On this basis a pure heat cycle is developed from the working cycle, as the calculations from actual figures and the graphical representation clearly illustrate. The reason this point has no particular importance in the theory of the reciprocating gas engine in practical operation is solely because, in the reciprocating engine, it is necessary, for purely mechanical reasons, that the temperature of the walls be kept low, this being an essential condition, which, if not fulfilled, prevents the engine running at all. Moreover, a consideration of the drop with relatively high temperatures is of too academic and abstract a nature even for the theorist.

From this point of view, the diagram resulting from the gas turbine with continuous combustion is also unfavourable. The combustion chamber is constantly filled with burnt gases, and continuously supplied with the mixed live gases; immediately after their admission they are raised to the ignition temperature, so that T_0 is very high, and the top temperature T_1 is therefore maintained at a high figure. If it be not so high as would be the case under otherwise similar circumstances but with explosion, when c_p and not c_v comes into consideration, then the greater portion of the heat set free would be radiated by the walls (Q_w being proportional to the 4th power of T with constant temperature).

It is therefore no longer a mechanical cycle but a pure heat process, which is better carried out in more suitable apparatus (boiler firing) than in appliances constructed for mechanical operation.

If, as has already been proposed, the coal is gasified in a generator under pressure, and combustion of the gas takes place continuously under this constant pressure (either in the generator or in another vessel), then, for the same reason as before, the greater part of the heat developed passes to the walls; it is always more of a heat process than a mechanical cycle.

An intermittent explosion process with the lowest possible temperature previous to ignition (therefore with air scavenging) and the largest possible nozzle opening (with nozzle valves) is the only possible one for an economical gas turbine. Moreover, this process is the most simple one for a gas turbine from a mechanical point of view.

6. CHARGING PROCESS.

After scavenging, the combustion chamber is filled with pure air in the condition $p_1v_1T_1$. Pure gas or a rich mixture of gas and air is blown in, pure gas being preferable, as explosion is then quite impossible in the admission valves, etc. The gas is drawn from a reservoir maintained under constant conditions $p_2v_2T_2$. If, as before, V is the capacity of the combustion chamber, then at a particular moment during the charging

$$\mathrm{G}v=\mathrm{V}$$

$$\mathrm{G}dv-vd\mathrm{G}=0$$

$$vd\mathrm{G}=\mathrm{F}cdz$$

$$c=\frac{\sqrt{2g}}{\phi}\sqrt{\mathrm{H}}$$

$$\mathrm{H}=\int_{p}^{p_2}vd\mathrm{P}$$

$$=\sqrt{2gv(p_2-p)}\text{ approximately.}$$

$$\left(\frac{\mathrm{V}+\int vd\mathrm{G}}{\mathrm{V}}\right)^{1\cdot 41}=\frac{p}{p_1}$$

when the change takes place adiabatically.

$$pv=\mathrm{RT}$$

$$\mathrm{Q}_w=\int \mathrm{T}d\mathrm{S}=\mathrm{Q}_r+\mathrm{Q}_c.$$

The heat taken by the walls $=\mathrm{Q}_w=\mathrm{Q}_r+\mathrm{Q}_c$.

During the admission of the charge, G, v, F, c, H, and T vary, and therefore the easiest method of arriving at the result is by the interpolation of definite intermediate values. The process is shown generally by the curves in fig. 11.

Naturally, heat is transmitted from the chamber walls to the gas mixture immediately there is a noticeable difference of temperature between them. If the temperature of the walls be high, this heat transference has an important effect, as is demonstrated in the general remarks on the combustion process. This is clearly shown in the calculations of the particular cases which follow.

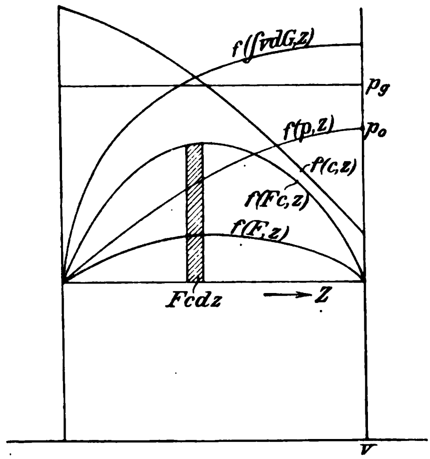


FIG. 11.—Curves showing charging process.

It is further to be noticed that a fall of temperature accompanies the expansion of the gases from the gas reservoir to the expansion chamber, but that the temperature rises with the compression of the mixture in the chamber.

The charging process can best be followed and examined from the entropy diagram. The character of the curve $f(F, z)$ (the gas valve lift curve) depends on the degree and completeness of the mixing of the gas with air, combined with the period of admission. For instance, a long period of admission and small valve lift give more complete and intimate mixing than a short time of admission and large valve lift. This point, of course, has some influence on the design and construction of the gas turbine.

Practical experience has shown that a more intimate mixing is obtained if the nozzle valve is slightly eased during the charging period, and this practical consideration naturally makes the theoretical examination of the charging process more complicated. In one direction this method is disadvantageous; it allows some air to pass out of the chamber, and a smaller quantity of gas, and hence the pressure p does not increase so rapidly as it otherwise would. The mixture is also richer, the temperature T is higher, and the radiation loss greater; but all these disadvantages can by proper arrangements be made of little significance, compared with the advantage of a better mixing.

It would, however, be a very serious mistake to deduce from this practical point, that the nozzle valve is in itself of little importance, and that the loss of a considerable amount of unburnt gases would be prevented, since the outlet section of the nozzle may be made relatively constricted. In the first place, the scavenge air has also to pass through the nozzle. From the point of view of economy, the pressure drop of the scavenge air should be as low as possible in its passage through the valve, since the energy required for this process must be provided by the gas turbine. Moreover, there is the further limitation due to the fact that a certain power has to be developed from each combustion chamber. The periods for cooling are the longest in the cycle, and must not be greatly prolonged.

The burnt gases must also convert their energy in the nozzles as quickly as possible, or too much heat is lost in the walls and the mechanical process develops into a heat cycle, in which either the cooling medium surrounding the walls or the mixture before ignition becomes heated, both effects being disadvantageous. In the gas turbine process, there should therefore be a large nozzle opening, and necessarily a means of closing the combustion chamber during admission of the charge, or a nozzle valve.

The intimacy and completeness of the mixture is considerably increased if the charging does not occur continuously but intermittently in short blasts. This is particularly important in the case of heavy liquid fuel. If this is injected in blasts by means of compressed air, the combustion chamber, which is relatively large compared with the reciprocating engine, is filled with alternate layers of pure air and gas and air, and a very complete combustion is thus obtainable.

7. EXPLOSION WAVE.

Physical chemistry explains a very important fact to designers of combustion and explosion machines.

Supposing a long pipe be filled with explosive carbon monoxide, and that the closed pipe be provided with an ignition apparatus at one end. If the gases be brought to a temperature of 650° C. at the one end by means of the igniter, combustion of the gas will take place at this end, and the pressure in the pipe will rise. The heat stream from the layers of gas which are first burned is propagated with a velocity, due to the ignition, of about 2·2 metres per second (for explosive gas $2H_2+O_2$, 35 metres per second), and brings the next layer of gas to the point of combustion. The pressure in the pipe is thus still further raised, and this increase takes place in the pipe with the velocity of sound, which is much higher than the speed of combustion. As the combustion passes along from layer to layer, the pressure continues to increase;

this is, in fact, the combustion pressure, which is given generally by the formula

$$\frac{p_1}{p_0} = \text{coefficient of contraction} \times \frac{T_1}{T_0},$$

where p_1 = pressure at end,
p_0 = pressure at beginning,
T_1 = temperature at end,
T_0 = temperature at beginning.

Considered from a purely theoretical standpoint, the pressure cannot rise higher in the pipe than exactly the combustion pressure after combustion of the total quantity of explosive gas; therefore, since for $2CO + O_2$

$$C_v = 0{\cdot}12 + 0{\cdot}000118T$$

$$T_1 = \frac{-0{\cdot}12 + \sqrt{2 \times 0{\cdot}000118(1550 + 0{\cdot}12T_0) + (0{\cdot}008118T_0)^2 + 0{\cdot}12^2}}{0{\cdot}000118}$$

$$\frac{p_1}{p_0} = 0{\cdot}67\frac{T}{T_0},$$

for $t_0 =$	0	100	200	300	400	500	600
„ $T_0 =$	273	373	473	573	673	773	873
„ $T_1 =$	4260	4270	4300	4310	4330	4370	4380
„ $\frac{p_1}{p_2} =$	10·45	7·66	6·1	5·02	4·32	3·78	3·36.

If, however, during the ignition the temperature of the unburnt gases rises to such an extent that spontaneous ignition occurs, the ignition from layer to layer will take place irregularly, and may lead to a bigger explosion if at a particular moment the unburnt mixed gases be brought suddenly and simultaneously to the point of combustion.

The temperature of the unburnt gas layers depends on the absorption from the sides of the walls, and above all on the pressure rise, following on the continuous combustion. The equation for the relation of the temperature rise to the increase of pressure is

$$\frac{T_1}{T_0} = \left(\frac{p_1}{p_0}\right)^{1-x'}.$$

For carbon monoxide the flash-point is 923° C. absolute.

Substituting this for T_1, then

$$(p_x : p_0)^{\cdot 41} = (p_1 : p_0)^{\cdot 41} = \frac{923}{T_0}.$$

If	$T_0 =$	273	373	473	573	673	873
and	$t_0 =$	0	100	200	300	400	500
then	$p_x : p_0 =$ 19·5	9·12	5·105	3·191	2·04	1·495	1·136.

Assuming that at a particular moment all the particles of unburnt gas are brought to the point of ignition, 923° C. absolute, and calling the pressure in this condition p_x = critical pressure, then the rise of pressure corresponding to this simultaneous explosion is

$$\frac{p}{p_x} = 0{\cdot}67 \times \frac{T_1}{923} = 0{\cdot}00064T_1$$

$$P = 0{\cdot}00064 T_1 p_x = \text{const.} \times p_0,$$

for	$t_0=$	0	100	200	300	400	500	600° C.
	Const. =	53·1	24·9	14·1	8·8	5·68	4·19	3·19.

Plotting the values of the pressures

$$p = \text{const.} \times p_0$$
$$p_x = \text{const.} \times p_0$$
$$p_1 = \text{const.} \times p_0$$

for temperatures from $t_0 = 0$ to 600° C., the curves shown in fig. 12 are obtained. It can be seen from these curves that with $t_0 = 150°$ C. the normal combustion pressure becomes equal to the critical pressure.

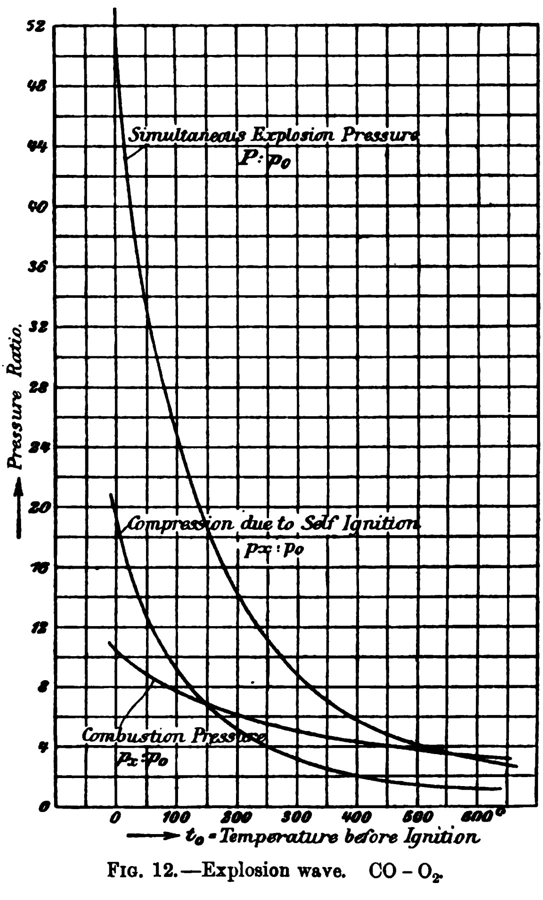


FIG. 12.—Explosion wave. $CO - O_2$.

In this condition all the unburnt gas is suddenly exploded, and the pressure increases to 18·8 times the initial pressure p_0, while the combustion pressure is only 6·8 times p_0. All the gas particles are now burnt with $t_0 = 150°$ C., so that the combustion pressure = the critical pressure. If t_0 is increased, however, beyond 150° C., the critical pressure becomes continuously less relative to the combustion pressure.

Above this limit, more gas remains unburnt, which is suddenly exploded at the moment in which the critical pressure is reached through ignition of a portion of the explosive gas. For instance, with $t_0 = 300°$ C. the critical pressure is 3·19 times p_0, and the combustion pressure 5·02 times p_0, so that after combustion of $\frac{3{\cdot}19}{5{\cdot}02} = 63{\cdot}5$ per cent. of the charge the critical pressure is reached; up to this moment the combustion pressure remains constant at $3{\cdot}19 p_0$, but immediately afterwards it increases to $8{\cdot}8 p_0$, or about 175 per cent. above the normal combustion pressure.

By a recoil of the combustion pressure wave, the pressure may rise locally to the critical pressure, and at this point there is a very marked increase of pressure. This has only been demonstrated up to now in physical chemistry, and as far as the author is aware the dependence of the explosion wave on the temperature before ignition, as described above, has not previously been considered.

These shocks occur at a lower temperature if the ignition temperature is lower, and the more hydrogen and methane there are in the mixture. G. Falk, in the *Annalen der Physik*, 1907 (24,449), gives the following ignition temperatures:—

$H_2 + O_2 + CO$	at	812° C.
$H_2 + O_2 + 2CO$	„	851° C.
$H_2 + O_2 + 4CO$	„	898° C.
$2H_2 + O_2 + 2CO$	„	877° C.
$2H_2 + O_2 + 4CO$	„	938° C.
$H_2 + 2O_2 + 2CO$	„	869° C.
$H_2 + 2O_2 + 4CO$	„	888° C.
$2H_2 + 3O_2 + 2CO$	„	825° C.

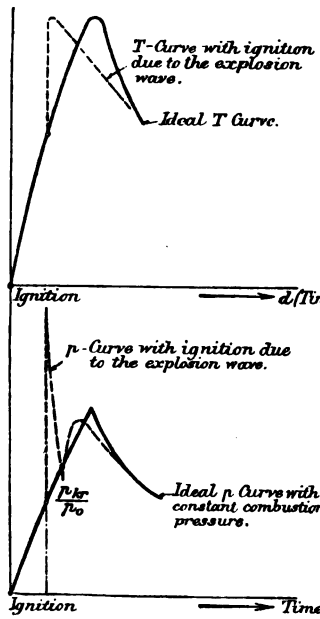


FIG. 13.—Explosion wave.

It follows from this explanation of the explosion impulses that:—

Any increase in the combustion pressure above the given value p_1 is deceptive and detrimental. It is deceptive inasmuch as it indicates momentarily a largely increased power; the effect of this power is, however, limited to a portion of the whole and is of short duration, and the useful work is no greater than with normal combustion of the whole charge.

The rise is detrimental owing to the possibility of shock due to the power being developed locally.

The explosion wave can only be advantageous when the regular (normal) combustion is inefficient, owing to premature escape of unburnt gas with very slow ignition by stages.

Considering an annular combustion chamber of a gas turbine, in which the ignition takes place at one end and the nozzle valves are at the other end, the ignition of the mixture would proceed slowly to the nozzle valve, but the accompanying rise of pressure would take place very quickly. A portion of the charge would therefore pass through the nozzle valve unburnt. In this case an explosion wave would be advantageous only if the escape of unburnt gas was prevented. Hence, to prevent such escape, the combustion chamber should be made of approximately spherical form, the ignition should take place simultaneously at various points, and care should be taken that no recoil wave can arise which would choke the combustion.

The designer of combustion and explosion engines has therefore in most cases to see that these explosion waves do not occur; but if it seems that their production is advisable, in view of the disadvantages which attend regular (normal) combustion, the effect of the explosion wave should be kept within reasonable limits.

The gas turbine diagrams figs. 14 to 25 show ignition by explosion waves.

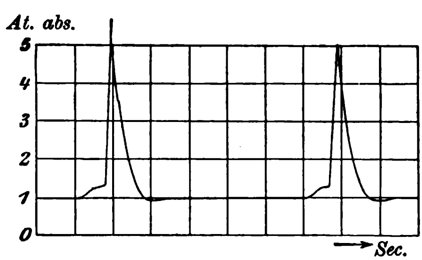


FIG. 14.

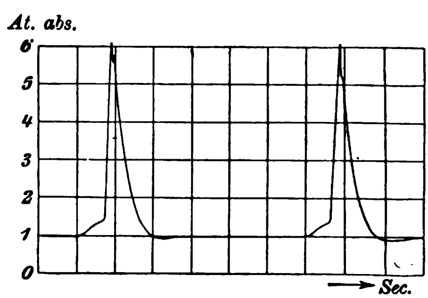


FIG. 15.

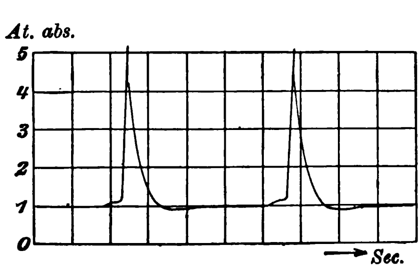


FIG. 16.

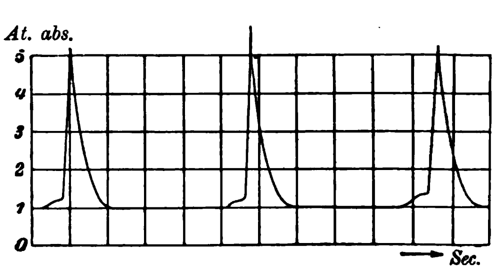


FIG. 17.

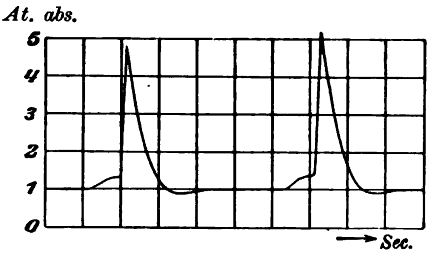


FIG. 18.

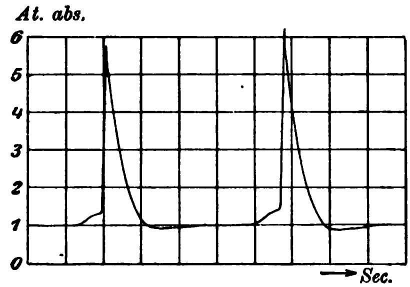


FIG. 19.

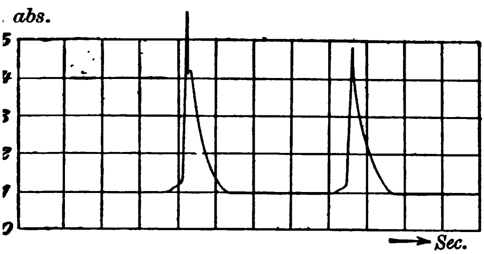


FIG. 20.

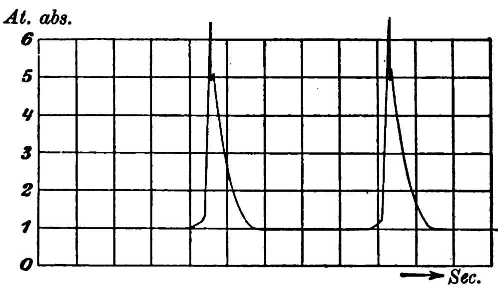


FIG. 21.

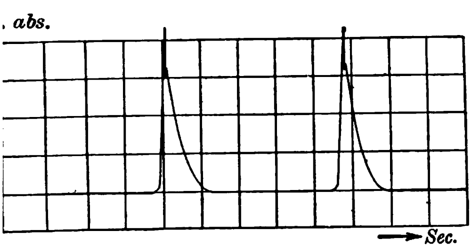


FIG. 22.

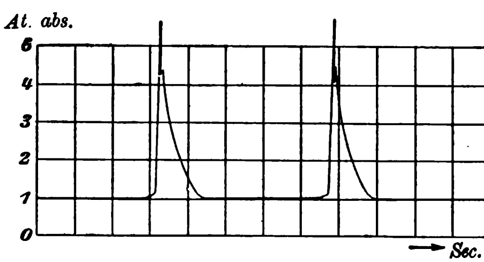


FIG. 23.

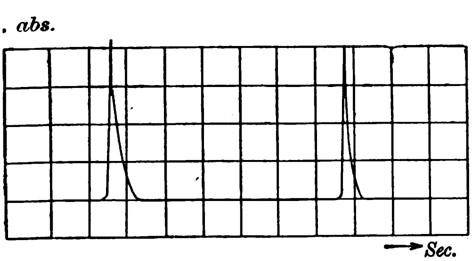


FIG. 24.

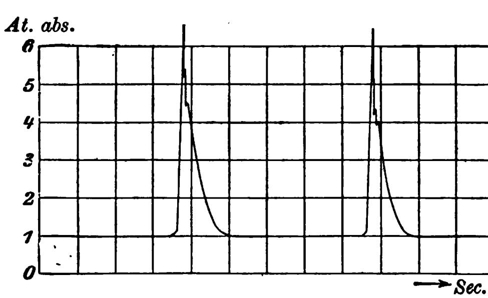


FIG. 25.

The diagrams figs. 26 to 29 show regular combustion from a particular point.

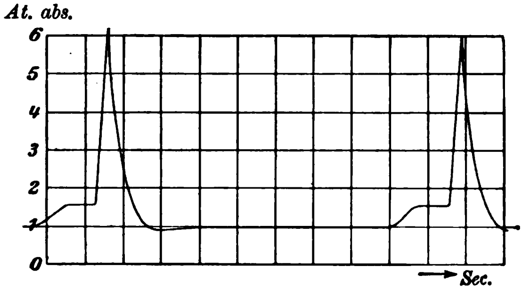


FIG. 26.

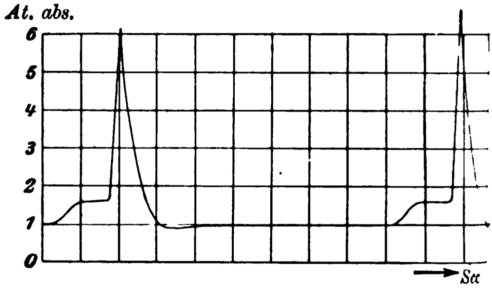


FIG. 27.

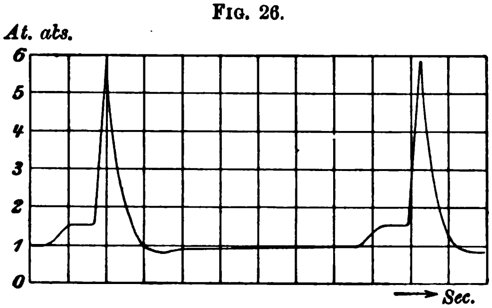


FIG. 28.

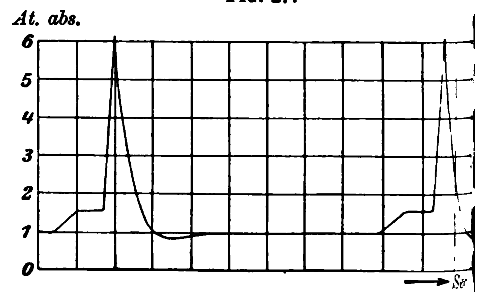


FIG. 29.

Figs. 30 to 39 show regular combustion at four points.

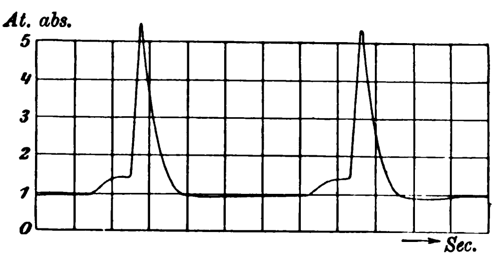


FIG. 30.

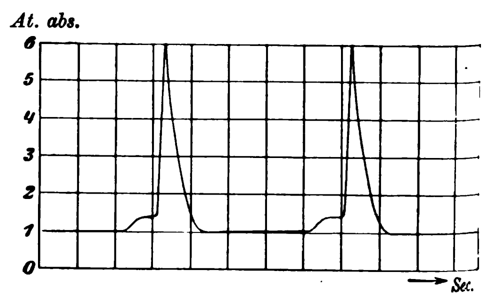


FIG. 31.

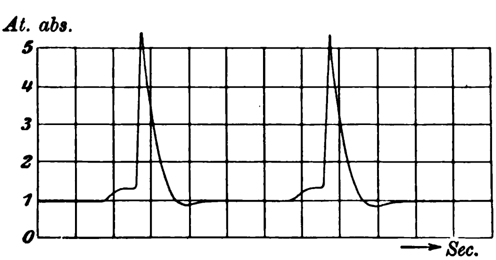


FIG. 32.

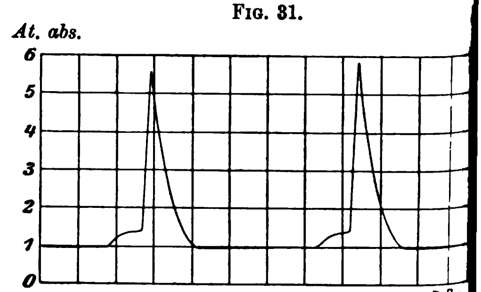


FIG. 33.

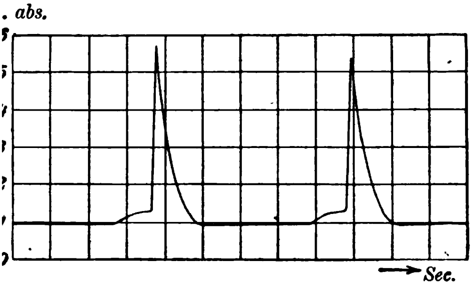


FIG. 34.

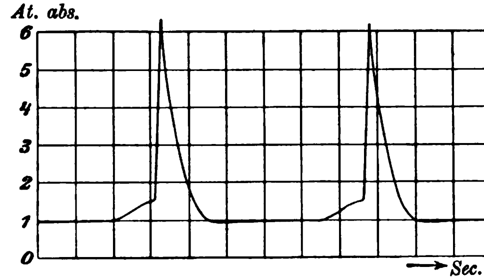


FIG. 35.

FIG. 36.

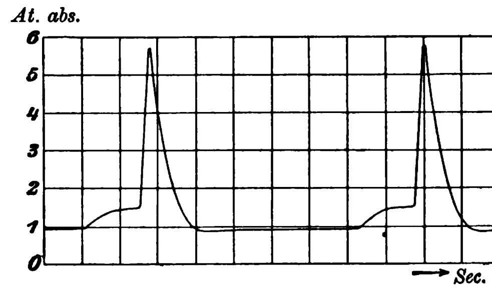


FIG. 37.

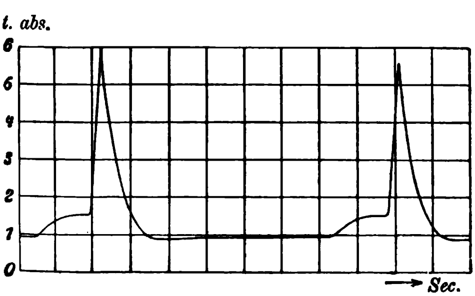


FIG. 38.

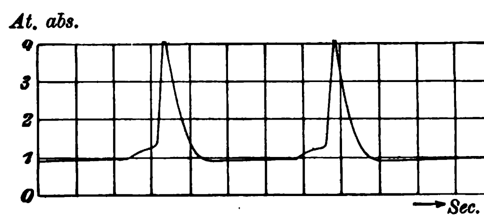


FIG. 39.

The calculations are made later for two examples showing the influence of T_0 on the process, the examples being taken from tests. In fig. 40 the values are indicated for the ratio $p_x : p_0$, assuming the ignition temperature $T_x = 923°$ C. abs., and also with $T_x = 858°$ C. abs. The corresponding values are:—

for $T_x = 923$	$t_0 =$	0	100	200	300	400	500	600° C.
	$p_x : p_0 =$	19·5	9·12	5·105	3·19	2·04	1·495	1·136
for $T_x = 858$	$p_x : p_0 =$	16·25	7·64	4·29	2·68	1·79	1·29	

From the examples on pages 33 and 37, the following values are also taken:—

G = 0·0287 kg.	$p_1 : p_0$ = 2·98	2·87	2·73	2·62	2·52	2·43
G = 0·0528 kg.	$p_1 : p_0$ = 4·06	3·8	3·61	3·37	3·19	3·05

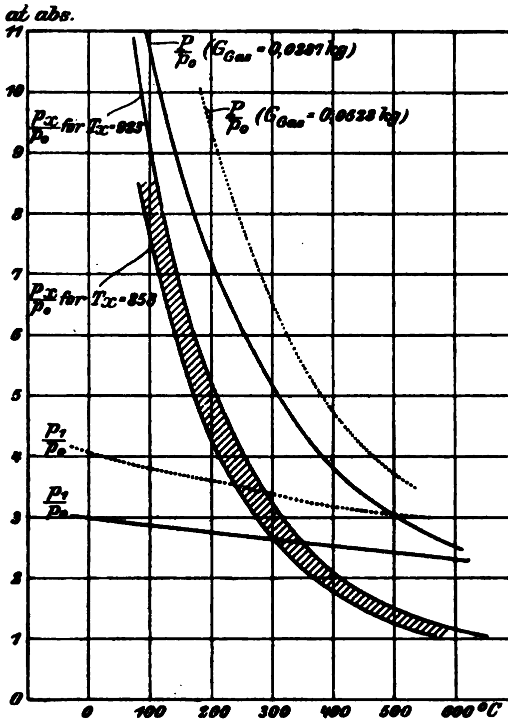


Fig. 40.—Explosion wave—power gas.

These values are also shown in the curves. Finally, the (sudden) explosion pressure for ignition temperature T = 923° can be calculated from the two examples mentioned, as

$$P = \text{coefficient contraction} \times \frac{T_1}{923} \times p_x.$$

For the first example, G = 0·0287 kg.,

$$P : p_0 = 17{\cdot}15 \quad 10{\cdot}6 \quad 7{\cdot}14 \quad 5{\cdot}17 \quad 3{\cdot}76 \quad 3{\cdot}06 \quad 2{\cdot}5;$$

and for the second example, G = 0·0528 kg.,

$$P : p_0 = 23{\cdot}2 \quad 13{\cdot}8 \quad 9{\cdot}3 \quad 6{\cdot}57 \quad 4{\cdot}7 \quad 3{\cdot}75.$$

These values are shown in the curves. It follows that, with the smaller charge of gas, the critical spontaneous ignition pressure is exceeded with a temperature of $t_0 > 350°$ C. if the ignition temperature is 923° C. The (critical) spontaneous ignition pressure is exceeded with a temperature of $t_0 > 300°$ C. if the ignition temperature of the mixture is 858° C. absolute. In any case, with this charge of gas, explosion waves will result when the temperature of 300–350° C. (before ignition) is exceeded. With the larger charge of gas the corresponding critical temperature zone is 230–285° C. The noteworthy result is deduced from this, that with the same temperature t_0 before ignition the explosion wave occurs earlier and with greater force, according as the charge of gas is increased. If, for example, the pressure in the gas reservoir is suddenly raised while the gas turbine is running, the explosion wave may rise even if the temperature t_0 remains unaltered. This is a proof that the temperature is in or about the critical zone, or that this has already been exceeded. The occurrence of the explosion wave is therefore a means of establishing the temperature t_0 previous to ignition. The direct experimental determination of this is very difficult, and can scarcely be accurately made.

The theoretical path of $p_1 : p_0$ is shown in the curves on fig. 41, whilst fig. 42 shows an interesting time-pressure diagram relative to the same.

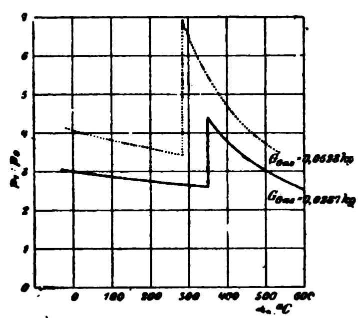


FIG. 41.—Explosion wave for power gas. Time-pressure diagram.

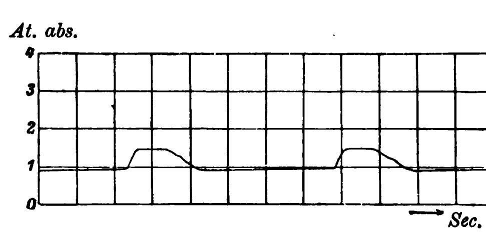


FIG. 42.—Experimental time-pressure diagram showing spontaneous ignition.

For the first particles of gas to ignite as they enter the combustion chamber, the temperature must have been $T_0 > 858$ to 923° C. absolute. Immediately after the external gas pressure is reached, the flow of gas into the chamber ceases, and then with pure gas the ignition cannot take place again.

It may be mentioned that the indicator diagrams taken agree very well with the calculations and theory developed above.

The theoretically and experimentally proved production of the explosion wave must always be taken into consideration. Care must be taken that an explosion wave developed during gas admission does not extend to the gas reservoir, not on account of danger of ignition of the gases contained therein (which is prevented by the use of gases not containing free oxygen), but on account of the rise of pressure with which it is attended. This can be effected by having gas inlet valves which close automatically when a certain back pressure is reached.

8. INFLUENCE OF TEMPERATURE PRIOR TO IGNITION. MECHANICAL AND HEAT CYCLES—TWO EXAMPLES.

In order to bring out more clearly the influence of the temperature T_0 before ignition, two examples are given, taken from experiments on the gas turbine. These examples will help to show more definitely the meaning of the foregoing remarks and formulæ. The gas turbine ran on power gas, the volumetric analysis at 15° C. and 1 atmosphere pressure (abs.) being

CO_2	5·8 per cent.
H_2	15·2 „
CO	25·8 „

The calorific value was 1179 calories. The indicated pressure in the chamber before ignition was $p_0 = 1{\cdot}22$ atmos. abs.; $p_2 = 0{\cdot}9$ atmos. abs.

In each charge 0·0287 kg. of gas was used, and the capacity of the chamber was V = 0·2 cub. metre.

Calculations will be made for the complete cycle with

$$t_0 = 0 \quad 100 \quad 200 \quad 300 \quad 400 \quad 500 \quad 600.$$

The analytical calculations are based on Langen's values.

For 1 kg. gas

	$c_v =$	$c_p =$
CO_2	0·12 + 0·000118T	0·165 + 0·000118T
H_2O	0·263 + 0·000239T	0·374 + 0·000239T
N	0·159 + 0·000043T	0·230 + 0·000043T
O	0·140 + 0·000037T	0·202 + 0·000037T
Air	0·155 + 0·000041T	0·224 + 0·000041T

For the entropy diagram Stodola's is used, with the values of Mallard and Lechatelier, in default of one for these values of Langen's. This entropy diagram will, however, only be used to determine the relative values; the absolute values will be obtained from the above figures of Langen, and for this reason the employment of the Mallard-Lechatelier diagram is not misleading.

TABLE I.

0·0287 kg. Gas.

$t_0 =$	0	100	200	300	400	500	600° C.
Volume of gas charge in condition $p_0v_0T_0$ . . . . . litres	20·8	28·5	36·1	43·7	51·4	59·0	66·6
Volume of air charge in condition $p_0v_0T_0$ . . . . . litres	179·2	171·5	163·9	156·3	148·6	141·0	133·4
Weight of air charge in condition $p_0v_0T_0$ . . . . . . kg.	0·283	0·198	0·149	0·1178	0·0951	0·0787	0·0658
Weight of total charge . . kg.	0·312	0·227	0·178	0·146	0·124	0·107	0·094
Heat supplied per 1 kg. charge calories	101·9	140·0	179·0	216·0	256·0	295·0	335·0
Ratio of mixture by weight . . .	1 : 9·85	1 : 6·9	1 : 5·19	1 : 4·1	1 : 3·31	1 : 2·73	1 : 2·29
Variable b of the specific heat per kg. .	0·000048	0·000050	0·000053	0·000055	0·000058	0·000061	0·000063
,, a_v ,, ,, ,,	0·156	0·156	0·156	0·156	0·156	0·156	0·156
Variable b' of the specific heat per kg.-molecule	0·00254	0·00254	0·00254	0·00261	...	...	...
Variable b of the specific heat per kg.-molecule	0·00291	0·00307	0·00323	0·0034	0·00355	0·00371	0·00386
Variable $a_v = a_v'$ of the specific heat per kg.-molecule	4·12	4·12	4·13	...	...	...	...
Coefficient of contraction . per cent.	98·0	97·2	96·4	95·7	94·9	94·2	93·4
T_1 ; formula p. 4 . . T_1°C. abs.	830·0	1100·0	1340·0	1570·0	1790·0	2000·0	2180·0
p_1 ,, p. 4 . . p_1 at abs.	3·64	3·50	3·33	3·18	3·07	2·97	2·85
ϵ ,, p. 6 } nearest to adiabatic	2·8	2·76	2·7	2·65	2·62	2·58	2·52
T_2 ,, p. 6 } expansion. T_2° C. abs.	572·0	780·0	980·0	1180·0	1370·0	1570·0	1735·0
$Q_w' + Q_w''$* . . calories per kg.	2·95	13·8	43·0	110·5	223·0	462·0	...
$Q_1 = Q + \int_{273}^{T_0} (a^{v'} + b'T)dT$,, ,,	101·9	157·2	214·0	269·8	329·3	389·0	450·0
Q_a 427 m.kg.	18850·0	24400·0	25900·0	21700·0	12650·0	...	...
Q per cent.	100·0	100·0	100·0	100·0	100·0	100·0	100·0
$\int_{273}^{T_0} (a_v' + b'T)dT$,,	0·0	10·6	17·1	20·0	22·0	...	...
Q_a ,,	43·3	40·8	33·9	23·6	11·58	...	...
Q_e ,,	53·8	60·2	59·6	47·5	27·7	...	...
$Q_r' + Q_r''$,,	2·9	9·7	23·6	48·8	83·0	...	...
η_k ,,	43·3	40·8	33·9	23·6	11·58	...	...
† η ,,	46·0	50·0	50·9	48·0	39·5	...	...
η_{total} ,,	19·9	20·4	17·3	11·3	4·57	...	...

* In this calculation no account is taken of the heat transmission by contact, first, because its determination is uncertain owing to the varying coefficient, and further, its value is small compared with the transmission by radiation.

† The efficiency is taken from curve on fig. 43.

$Q_w' + Q_w''$ are joined together and the expansion period assumed. The calculation of $Q_w' + Q_w''$ was obtained from the p-time diagram by dividing the expansion line into three sections of equal pressure.

$$Q_w' + Q_w'' = C_1 F z \left[\left(\frac{E}{100} \right)^4 - \left(\frac{T}{100} \right)^4 \right].$$

$$F = 1 \cdot 25 \text{ sq. m.,}$$
$$za = 0 \cdot 000077.$$
$$C_1 = 4.$$

$$Q_w' + Q_w'' = 0 \cdot 000389 \left[\left(\frac{T_1 T_2}{10{,}000} \right)^2 - \left(\frac{T}{100} \right)^4 \right].$$

For T, $T_0 + 50$ may be substituted.

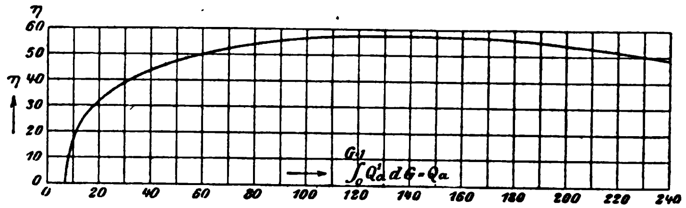


FIG. 43.—Curve of η (Table I.).

The entropy diagram is in conformity.

For $t_0 = 400°$ C.

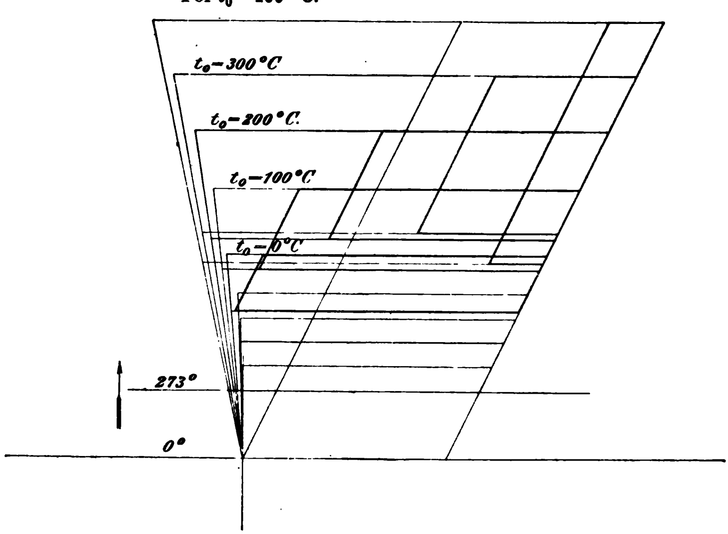


FIG. 44.—Entropy diagram for Table I.

The representation of the final values of the above table in a curve gives the diagram fig. 45.*

If the heat supplied (Q) remains constant, and also p_0 and p_2, the efficiency of combustion decreases very rapidly as T_0 increases, owing to the heat transmitted through radiation. If t_0 exceeds 400° C., the cycle is no longer a mechanical one, but a pure heat cycle.

This is shown in fig. 46, where the various quantities of heat are represented in percentages of Q.

To one point particular attention may be paid. The foregoing examination shows that p_0 remains constant when t_0 varies, and this

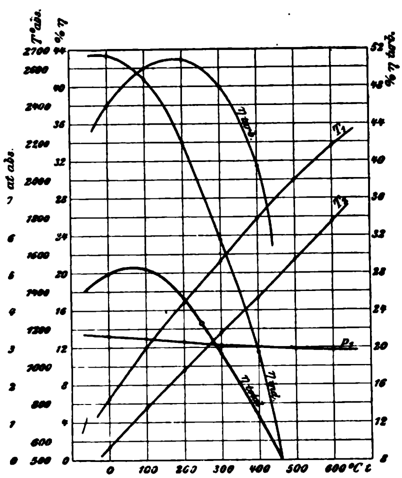


FIG. 45.—Efficiency curves (Table I.).

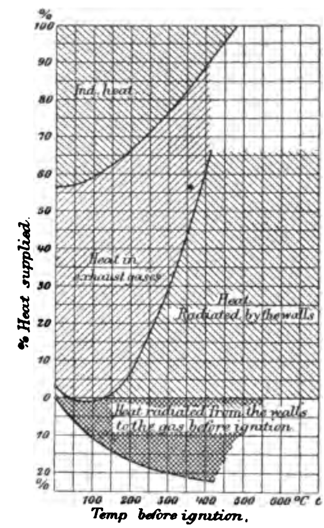


FIG. 46.—Heat balance (Table I.).

has a practical bearing, indicating that p_0 depends mainly only on the weight of gas supplied. If it were possible to so regulate the influence of the heat of the walls that during the charging of the combustible gas with air no heat transmission took place, but that after complete admission and before ignition the heat could be transmitted, and therefore the mixture would be brought from the temperature

$$\frac{lt_s+gt_g}{l+g}$$

to the temperature t_0, then with the pre-heating of the mixture a corresponding rise of pressure p_0 would occur. This would, of course, be

* In this and the following figures the same letters are used : $\eta_{turb.}$ with η ; $\eta_{ind.}$ with η_a ; "ind. heat" with "available heat" ; t' with t_0.

favourable to the cycle, but can only be accomplished by making the time of admission short, and the interval between the end of admission and ignition very great. The consequence of this, however, would be an exceedingly unfavourable one, due to the eddying in the mixture. The lighter gas would separate from the air of combustion and collect on the top. From a constructional standpoint also there would be difficulties, as there is little time at disposal. It must be recognised, therefore, as a fact which is proved in practice, that pre-heating of the mixture does not produce a corresponding rise of pressure.

The second example gives results on the relationships when a bigger charge of gas is supplied. Apart from this the conditions are the same.

$$p_0 = 1{\cdot}375 \text{ at. abs.}$$

Charge of gas weighs 0·8528 kg.

$$p_2 = 0{\cdot}925 \text{ at. abs.}$$

The corresponding curves are given in figs. 47, 48, and 49.

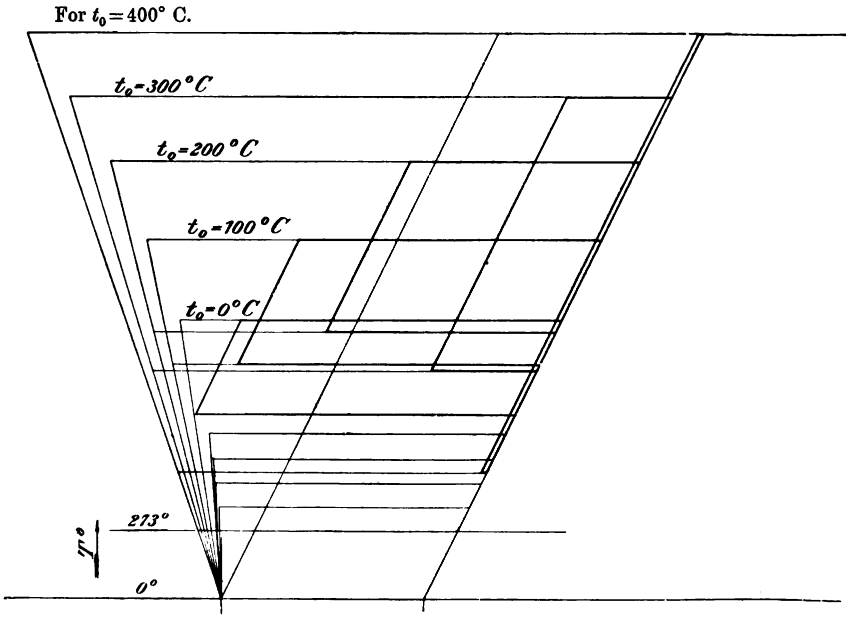


FIG. 47.—Entropy diagram for Table II.

TABLE II.

$G_{gas} = 0{\cdot}0528$ kg.

$t_0 =$	0	100	200	300	400	500	600° C.
Volume of gas charge in condition $p_0v_0T_0$ . . . . litres	34·0	46·3	58·8	71·4	83·8	96·1	108·5
Volume of air charge in condition $p_0v_0T_0$ . . . . litres	166·0	153·7	141·2	128·6	116·2	103·9	91·5
Weight of air charge in condition $p_0v_0T_0$ . . . . . . kg.	0·295	0·200	0·145	0·109	0·084	0·0655	0·051
Weight of total charge . . kg.	0·348	0·253	0·198	0·162	0·137	1·118	0·104
Heat supplied per 1 kg. charge calories	166·0	229·0	293·0	357·0	423·0	492·0	558·0
Ratio of mixture by weight . . .	1 : 5·6	1 : 3·79	1 : 2·75	1 : 2·07	1 : 1·59	1 : 1·24	1 : 0·967
Variable b of the specific heat per kg. .	0·000052	0·000056	0·000060	0·000064	0·000069	0·000073	...
,, a_v ,, ,, ,,	0·156	0·156	0·156	0·156	0·156	0·156	...
Variable b' of the specific heat per kg.-molecule before combustion . .	0·00257	0·00267	0·00273	0·00277	0·00287	...	...
Variable b of the specific heat per kg.-molecule after combustion . .	0·00321	0·00347	0·00374	0·00398	0·00419	...	...
Variable $a_v = a_v'$ of the specific heat per kg.	4·12	4·12	4·12	4·13	4·13	...	...
Coefficient of contraction . per cent.	96·5	95·5	94·0	93·0	92·5	91·6	...
T_1 ; formula p. 4 . $T_1°$ C. abs.	1135	1465	1785	2050	2300	2540	...
p_1 ,, p. 4 . . p_1 at abs.	5·5	5·15	4·88	4·57	4·32	4·13	...
ϵ ,, p. 6 } nearest to adiabatic	3·79	3·73	3·64	3·57	3 437	3·43	...
T_2 ,, p. 6 } expansion. $T_2°$ C. abs.	745	982	1230	1485	1715	1955	...
$Q_w' + Q_w''$ . . . calories per kg.	7·8	32	96	238	...	...	...
$Q_1 = Q + \int_{273}^{T_0} (a^{v\prime} + b'T)dT$,, ,,	166	246·4	329·2	411·9	...	...	...
Q_a 427 m.kg.	75·5	90·8	87·0	62·1	...	...	...
Q per cent.	100	100	100	100	...	...	...
$\int_{273}^{T_0} (a_v' + b'T)dT$,,	0·0	6·5	9·3	11·5	...	...	...
Q_a ,,	45·5	39·5	29·6	17·3	...	...	...
Q_e ,,	49·8	53·0	47·6	29·7	...	...	...
$Q' + Q_r'$,,	4·7	13·8	31·9	64·4	...	...	...
η_k ,,	45·5	39·5	29·6	17·3	..	...	...
η ,,	53·4	55·5	55·0	50·8	...	...	...
η_{total} ,,	24·3	21·9	16·3	8·78	...	...	...

With the bigger charge the efficiency η_{total} rises higher than before, but falls off more rapidly. At a temperature $t_0 = 300°$ C. the cycle is more a heat cycle than a mechanical cycle.

These calculations, based on experimental data, lead us to the conclusion that it is of the greatest importance in a gas turbine to keep the radiation from the walls within reasonable limits; otherwise the desired mechanical cycle becomes a heat cycle, which is disadvantageous. To this end, the opening from the combustion chamber to the turbine (closed during the charging process) should be as large as possible. The cycle should also be carried out with the maximum temperature T_1 not too excessive, and therefore the temperature T_0 (temperature before ignition) as low as possible. The walls of the actual combustion chamber must be cooled, and the walls behind the nozzle valves insulated. The supply of scavenge air and gas must be at a low temperature.

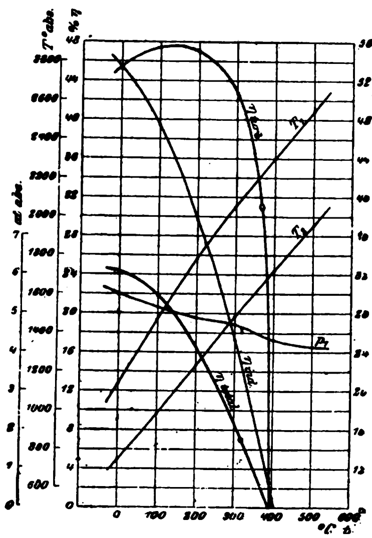


FIG. 48.—Curves from Table II.

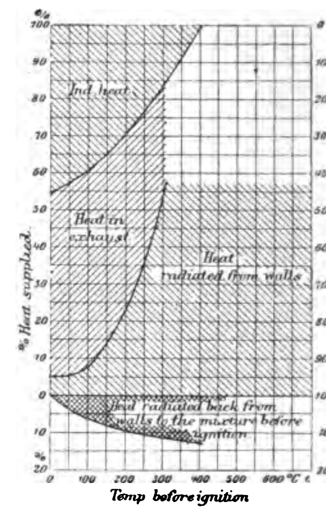


FIG. 49.—Heat balance. Table II.

9. INFLUENCE OF THE GAS CHARGE. EFFICIENCY CURVES. TWO EXAMPLES.

To illustrate the influence of the gas charge (or admission) two further examples will be worked out, for

$$t_0 = 15^\circ \text{ C.}$$

and

$$t_0 = 100^\circ \text{ C.}$$

Power gas was employed as before. The capacity of the combustion chamber is $V = 200$ litres, and the final pressure in both cases is

$$p_2 = 0{\cdot}9 \text{ at. abs.}$$

The data given are the final results from the individual cases, and based on practical tests.

TABLE III.

$t_0 = 15°$ C.

Weight of gas supplied per charge . kg.	0·030	0·050	0·070	0·090	0·110
Pressure p_0 immediately before ignition at. abs.	1·23	1·39	1·52	1·67	1·83
Volume of gas at 15° C. v_{0g} . litres	22·8	33·7	43·0	50·2	56·2
Volume of air at 15° C. . . ,,	177·2	166·3	157·0	149·8	143·8
Weight of air . . . . . . kg.	0·267	0·282	0·292	0·305	0·329
Weight of charge . . . G kg.	0·297	0·332	0·362	0·395	0·439
Heat supplied, Q . . calories per kg.	119	177	228	268	295
Mixture ratio by weight	1 : 8·9	1 : 5·64	1 : 4·17	1 : 3·39	1 : 2·99
Variable b of the specific heat per kg. .	0·0000485	0·0000516	0·000055	0·000058	0·000059
,, a_v ,, ,, ,,	0·156	0·156	0·156	0·156	0·156
,, a_p ,, ,, ,,	0·225	0·225	0·225	0·225	0·225
Coefficient of contraction . . per cent.	98·0	96·5	95·8	95·0	94·5
T_1 from formula p. 4 . . °C. abs.	1112·0	1195·0	1398·0	1560·0	1670·0
p_1 ,, ,, p. 4 . . at. abs.	4·63	5·56	7·08	8·6	10·0
T_2 ,, ,, p. 6 Entropy diagram °C. abs.	775·0	800·0	930·0	965·0	1000·0
$Q_w' + Q_w''$	9·55	10·6	17·5	22·1	24·6
Q_a	68·9	82·8	105·5	139·2	134·6
η_v per cent.	44·5	46·6	46·2	52·0	45·6
η ,,	52·0	54·6	57·0	58·0	58·0
η_{total} ,,	23·2	25·4	26·3	30·1	26·4

TABLE IV.

$t_0 = 100°$ C.

Weight of gas supplied per charge . kg.	0·030	0·050	0·070	0·90	0·110
Pressure p_0 immediately before ignition at. abs.	1·23	1·39	1·52	1·67	1·83
Volume of gas at 15° C. v_{0g} . . litres	29·4	43·0	55·2	65·0	72·6
Volume of air at 15° C. . . . . ,,	170·6	157·0	144·2	135·0	127·4
Weight of air . . . . . . kg.	199·5	207·0	208·0	214·0	220·0
Weight of charge . . . G kg.	229·5	257·0	278·0	304·0	330·0
Heat supplied, Q . . calories per kg.	154·0	229·0	296·0	349·0	392·0
Mixture ratio by weight	1 : 5·69	1 : 4·14	1 : 2·97	1 : 2·38	1 : 2
Variable b of the specific heat per kg. . .	0·000052	0·000055	0·000059	0·000063	0·000065
,, a_v ,, ,, ,, . .	0·156	0·156	0·156	0·156	0·156
,, a_p ,, ,, ,, . .	0·225	0·225	0·225	0·225	0·225
Coefficient of contraction . . per cent.	96·5	95·6	94·1	93·5	93·0
T_1 from formula p. 4 . . . ° C. abs.	1157·0	1470·0	1730·0	1920·0	2050·0
p_1 ,, ,, p. 4 . . . at. abs.	3·67	5·22	6·65	8·02	9·35
T_2 ,, ,, p. 6 Entropy diagram ° C. abs.	814·0	982·0	1142·0	1266·0	1250·0
$Q_w' + Q_w''$ calories per kg.	15·4	31·1	54·2	75·0	77·8
Q_a ,,	58·7	90·1	116·0	138·2	162·0
η_v per cent.	37·8	36·6	37·0	37·6	39·5
η ,,	50·0	55·5	57·7	58·0	57·2
η_{total} ,,	18·9	20·3	21·4	21·8	22·6

The calculations of the final results are used in the curves in fig. 50.

The results are plotted on the following curves, as the first example in fig. 51.

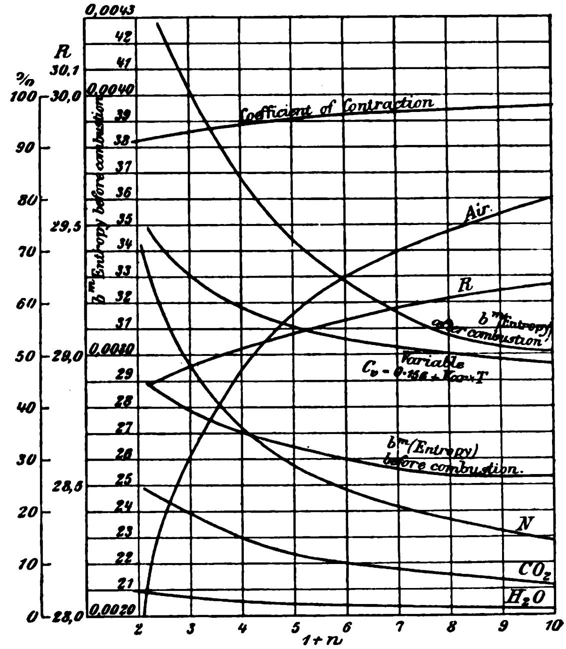


FIG. 50.—Curves for deriving final results.

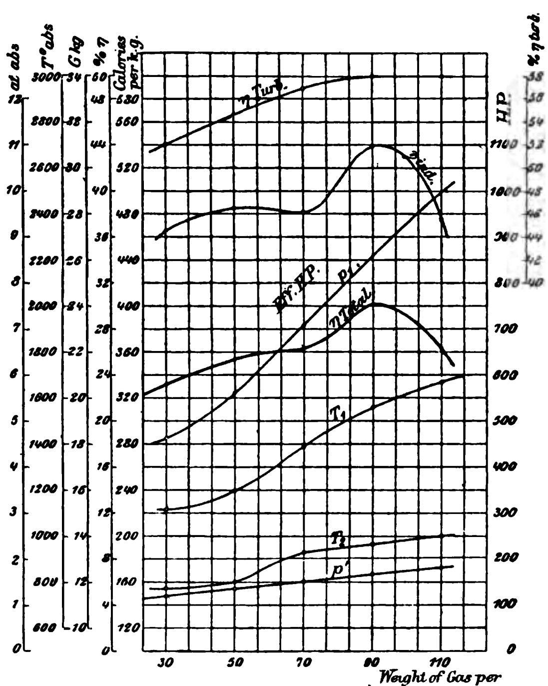


FIG. 51.—Curves from Table III.

The second example is plotted in fig. 52.
The entropy diagrams are given in figs. 53 and 54.

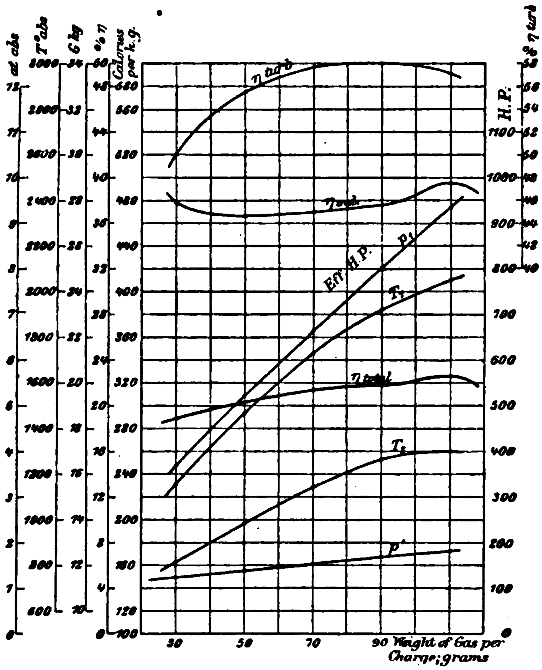


FIG. 52.—Curves from Table IV.

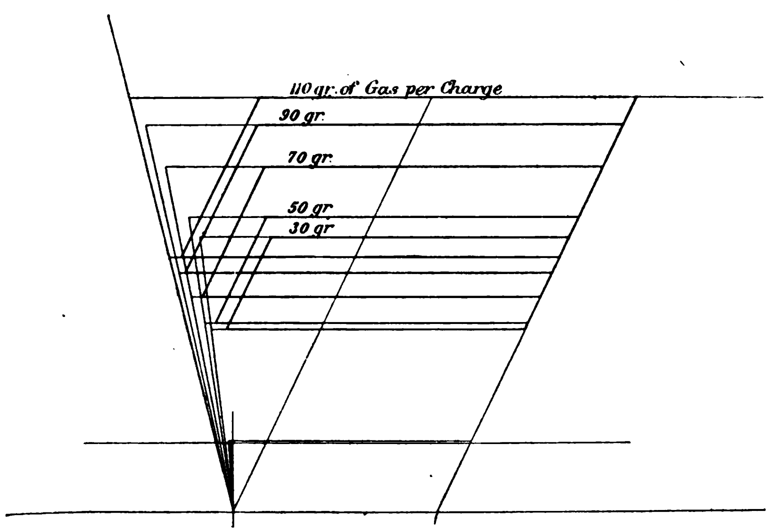


FIG. 53.—Entropy diagram from Table III.

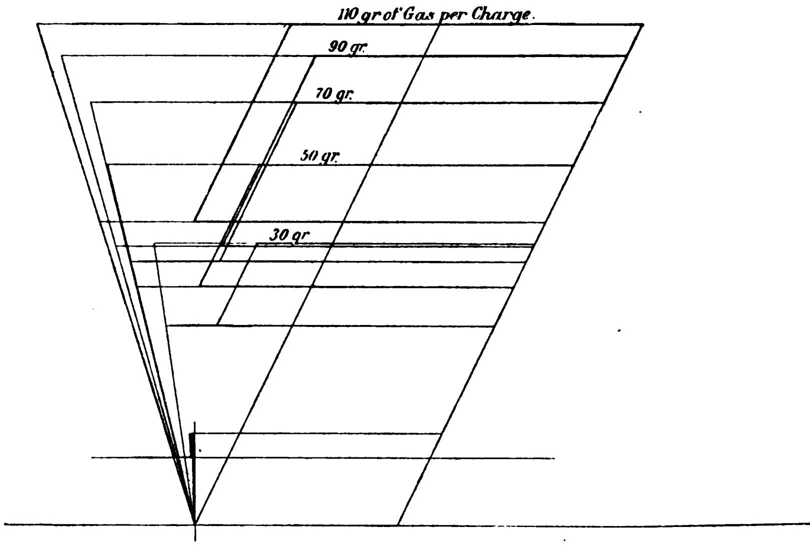


FIG. 54.—Entropy diagram from Table IV.

The results are further represented in fig. 55 in curves with three axes, namely:—

Over-all efficiency.
t_0, temperature before ignition.
G, charge of gas.

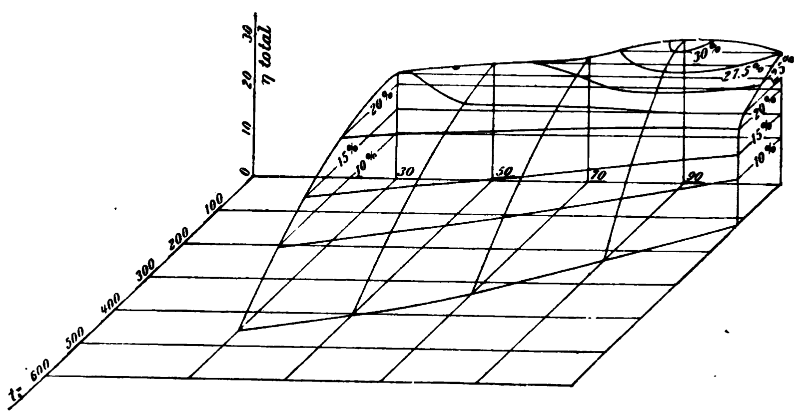


FIG. 55.—Curves from Tables III. and IV.

From the representation with three axes, the factors influencing the efficiency are clearly shown. The over-all efficiency is drawn at right angles to the plane $(G_g - t_0)$. The surface, in which the end points of the values of over-all efficiency lie, is intersected by horizontal planes for 10, 15, 20, 22·5, 25, 27·5 and 30 per cent. The curves of intersection so formed are drawn, and also the curves of intersection of the efficiency plane, with planes parallel to the $(\eta_{total} - t_0)$ plane. It is evident from the diagram that the efficiency plane drops rapidly towards the front in the direction of the increasing temperature t_0 (before ignition), and the bigger the charge of gas, the steeper is the drop. The efficiency plane touches its highest point in the neighbourhood of 0° C., and reaches its maximum value of 30 per cent. with a charge of gas of about 90 grams. If the efficiency plane be prolonged behind the $(\eta_{total} - G_{gas})$ plane, *i.e.* for values of t_0 considerably less than 0° C., the plane falls away again, to a greater or lesser extent according as the external temperature is higher or lower.

10. INFLUENCE OF THE SIZE OF THE COMBUSTION CHAMBER. EXAMPLES.

The foregoing examples are based on experiments with our second and larger gas turbine, having combustion chambers of 200 litres capacity, and driven with power gas. The experiments which will now be examined on the same lines show the influence of the size of the chambers, and were made on our first turbine, having explosion chambers each of 50 litres capacity and working on lighting gas. The surface of each chamber was 0·500 sq. metre. The ratio of the volume to the surface in the previous examples was

$$\frac{V}{O} = \frac{0{\cdot}200}{1{\cdot}27} = 1 : 6{\cdot}35,$$

while in the following examples it was

$$\frac{V}{O} = \frac{0{\cdot}050}{0{\cdot}500} = 1 : 10.$$

As a comparison for this ratio, a reciprocating gas engine of 870 mm. cylinder diameter and 1000 mm. stroke may be used. In this, the clearance volume, internal and external, is

$$V_i = \text{Compression space in cubic metres},$$

$$V_a = V_i + {\cdot}87^2 \frac{\pi}{4} \times 1 = V_i \times {\cdot}592.$$

$$\frac{V_i + {\cdot}592}{V_i} = \text{Compression ratio} = 4{\cdot}4 \text{ approximately}.$$

$$V_i = {\cdot}174 \text{ cub. metre},$$
$$V_a = {\cdot}766 \text{ cub. metre}.$$

The corresponding values of the surfaces are

$$O_i = \frac{2 \times \pi \times \cdot 87^2}{4} + 0\cdot 87 \times \pi \times h_i,$$

$$h_i = \frac{0\cdot 174}{0\cdot 592} = 0\cdot 0293; \quad h_a = 1\cdot 0293;$$

therefore

$$O_i = 2 \times \cdot 592 + \cdot 0802 = 1\cdot 264 \text{ sq. metres},$$

$$O_a = 1\cdot 184 + 2\cdot 82 = 4\cdot 004 \text{ sq. metres}.$$

The corresponding ratios are

$$\frac{V_i}{O_i} = \frac{0\cdot 174}{1\cdot 264} = 1 : 7\cdot 28,$$

$$\frac{V_a}{O_a} = \frac{0\cdot 766}{4\cdot 004} = 1 : 5\cdot 22.$$

The more favourable ratio is obtained from a combustion space of spherical form, and the less favourable ratio from the cylindrical form; a larger space gives a better ratio than a smaller one.

For the transmission by radiation we have, as before,

$$Q_r = C_1 F z \left[\left(\frac{E}{100}\right)^4 - \left(\frac{T}{100}\right)^4 \right]$$

$$= C_1 F z a \left[\left(\frac{T_1 T_2}{10{,}000}\right)^2 - \left(\frac{T}{100}\right)^4 \right].$$

In this case $C_1Fza = \cdot 000154$, and $p_0 = 1\cdot 12$ at. abs.

The charge of gas is $Q = 1\cdot 79$ grms.

Heat supplied per charge, $Q = 17\cdot 9$ calories.

The volumetric composition of the lighting gas at 0° C. and 760 mm. pressure was:—

$H_2 = 52$ per cent.
$CH_4 = 29$ „
$CO = 8$ „
$C_mH_m = 4$ „
$CO_2 = 2$ „
$N = 4\cdot 5$ „

The composition per kg. of gas was:—

	Vol. per cent.	γ	γv per cent.	Weight per cent.
H_2	52	0·09	0·0468	8·85
CH_4	29	0·716	0·207	39·1
CO	8	1·25	0·1	18·9
C_mH_m	4	2·00	0·08	15·1
CO_2	2	1·96	0·0392	7·4
N	4·5	1·254	0·0563	10·65
			0·5293	

$$\gamma = \cdot 53$$

The calorific value per kg. of gas was found to be about 10,000 calories.

To obtain the air required for combustion, proceed as follows:—

	Weight per cent.	Air required for 1 kg. of gas.	
H_2	8·85	33·35 kg. air	2·96 kg. air.
CH_4	39·1	16·77 „	6·55 „
CO	18·9	2·4 „	0·454 „
C_mH_m	15·1	14·0 „ (about)	2·12 „
CO_2	7·4		
N	10·65		
			12·08 kg. air.

1 kg. of gas requires 12·08 kg. of air for complete combustion.

The exhaust gases from 1 kg. of gas are as follows:—

$$CO_2 = \cdot 074 + \cdot 391 \times 2\cdot 745 + \cdot 189 \times 1\cdot 571 + \cdot 151 \times \cdot 31 = 1\cdot 908 \text{ kg.}$$
$$H_2O = \cdot 0885 \times 8\cdot 937 + \cdot 391 \times 2\cdot 248 + \cdot 151 \times 1\cdot 3 = 1\cdot 864 \text{ ,,}$$
$$N = \cdot 106 + 12\cdot 08 \times 0\cdot 762 = 9\cdot 336 \text{ ,,}$$

This assumes that the combustion takes place without an excess of air.

TABLE V.

Charge of gas = 0·00179 kg.					
	$t_0=$	0°	100°	200°	300°
Total volume of gas charge in condition $p_0v_0T_0$	litres	3·02	4·12	5·22	6·32
,, ,, air ,, ,, ,,	,,	46·98	45·88	44·78	43·68
Weight of air charge in condition $p_0v_0T_0$	kg.	0·066	0·0472	0·0363	0·0292
,, total ,, ,, ,,	,,	0·0678	0·0490	0·0381	0·0310
Heat supplied per kg. charge Q	calories	264·0	365·0	470·0	577·0
Mixture ratio by weight, 1 : *n*		1 : 36·9	1 : 26·4	1 : 20·3	1 : 16·3
1 + *n*		37·9	27·4	21·3	17·3
Exhaust Gases:					
Weight of surplus air per 1 kg. of gas	kg.	24·82	14·32	8·22	4·22
Weight of CO_2 per 1 kg. of gas	,,	1·908	1·908	1·908	1·908
,, H_2O ,, ,,	,,	1·864	1·864	1·864	1·864
,, N ,, ,,	,,	9·336	9·336	9·336	9·336
Weight of exhaust gases per 1 kg. of gas	per cent.	100·0	100·0	100·0	100·0
Weight of CO_2 per 1 kg. of gas	,,	5·04	6·96	8·96	11·8
,, H_2O ,, ,,	,,	4·92	6·8	8·76	10·8
,, N ,, ,,	,,	24·6	34·0	43·8	54·9
Weight of surplus air per 1 kg. of gas	,,	65·4	52·1	38·6	24·4
Variable a_v of specific heat per 1 kg. =					
surplus air = ·155 per cent.		0·1015	0·0808	0·0598	0·0379
CO_2 = ·120 ,,		0·00607	0·00837	0·0107	0·0133
H_2O = ·263 ,,		0·01295	0·0179	0·0231	0·0284
N = ·159 ,,		0·0391	0·054	0·0696	0·0858
a_v =		0·1606	0·1611	0·1632	0·1654

TABLE V.—*continued.*

Charge of gas = 0·00179 kg.				
$t_0=$	0°	100°	200°	300°
Variable b of specific heat per 1 kg. exhaust gases = surplus air = ·000041 per cent	0·0000268	0·0000214	0·0000158	0·0000101
CO_2 = ·000118 ,,	59	82	105	131
H_2O = ·000239 ,,	118	163	209	266
N = ·000043 ,,	106	145	189	232
$T_1=\dfrac{-a_v+\sqrt{a_v^2+2b(Q_1+273a_v)+(273b)^2}}{b}$ $b=$	0·0000551	0·0000605	0·0000661	0·0000730
Per 1 kg. charge $\int_{273}^{T_0}(a_v'+b'T)dT$	0	17·4	36·2	54·9
$Q_1=$	264	382·4	506·2	631·9
$273a_v=$	44·1	44·1	44·5	45·2
$Q_1+273a_v=$	308·1	426·5	550·7	677·1
$(273b)^2=$	0·000226	0·000273	0·000326	0·0004
$a_v^2=$	0·0257	0·0257	0·0261	0·0264
$2b(Q_1+273a_v)=$	0·034	0·0515	0·0728	0·0987
$a_v^2+2b(Q_1+273a_v)+(273b)^2=$	0·0599	0·0775	0·0992	0·1255
$X=\sqrt{a_v^2+2b(Q_1+273a_v)+(273b)^2}=$	0·244	0·278	0·315	0·354
$-a_v+\sqrt{X}=$	0·083	0·117	0·152	0·189
$T_1=$	1510	1930	2300	2580
Coefficient of contraction $\phi=$. per cent.	98·5	97·9	97·3	96·7
$p_1=p_0\dfrac{T}{T_0}\phi$. at. abs.	6·08	5·68	5·31	4·87
Entropy Diagram :				
Before combustion CO_2, mb'. per cent. ; $b'=0·0774$; $m=44$				
Permanent, gas mb'. per cent. ; $b'=0·0244$; $m=28$	0·0685	...	...	...
$\Sigma mb'$. per cent.	0·0685	...	...	...
$CO_2 m$. ,,				
Permanent gas m. ,,	28·0	...	...	...
Σm. ,,	28·0	...	...	...
$b'=$	0·00245	0·00245	0·00245	0·00245
After combustion CO_2, mb. per cent. ; $b=·00774$; $m=44$	0·0172	0·0237	0·0305	0·00378
H_2O, mb. ,, $b=·00574$; $m=18$	0·00509	0·00703	0·00903	0·0112
Permanent gas, mb. ,, $b=·00244$; $m=28$	0·0617	0·059	0·0564	0·0538
Σmb. per cent. =	0·0840	0·0897	0·0959	0·1028
CO_2 m. ,, =	2·22	3·06	3·94	4·87
H_2O m. ,, =	0·888	1·225	1·575	1·94
Permanent gas m. ,, =	25·2	24·2	23·1	22·0
Σm. ,, =	28·308	28·485	28·615	28·81
$b=$	0·00296	0·00315	0·00335	0·00357
From the entropy diagram for $Q_w'+Q_w''$ per 1 kg. charge =	45·3	137·0	281·0	394·0WE
$T_2=$	940·0	1080·0	1130·0	1100·0° C. abs.
$Q_a=$	96·0	96·0	100·5	70WE
$Q_e=$	123·0	149·0	126·0	165WE
Q per cent.	100·0	100·0	100·0	100
Q_r ,,	17·1	38·1	59·8	68·2
Q_a ,,	36·3	26·3	21·3	12·1
Q_e ,,	46·7	40·8	26·9	28·6
η_x	36·3	26·3	21 3	12·1
Single wheel η	28	28	26	30
η_{total} ,,	10·4	7·38	5·55	3·63

The main values from the above calculation are shown in the curves in fig. 56, and the series of entropy diagrams are given in fig. 57.

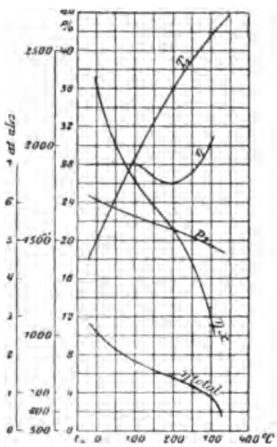


FIG. 56.—Curves from Table V.

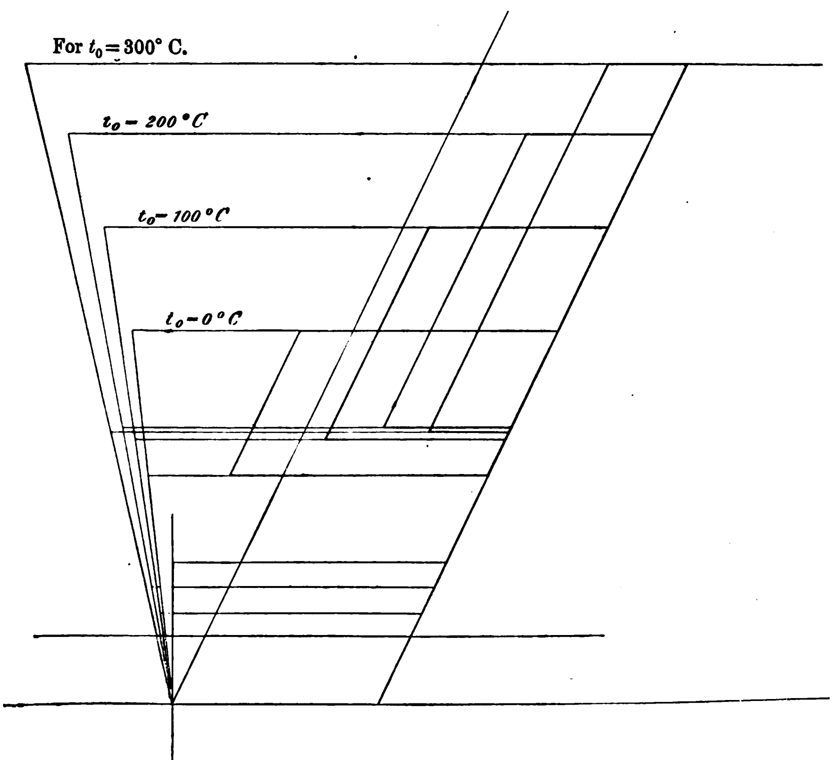


FIG. 57.—Entropy diagrams for Table V.

These calculations, which agree very well with the experiments, show that the combustion in a small combustion chamber, under otherwise similar conditions, gives less favourable results than in a larger combustion chamber. Comparing the curve η_x, which is the only one entering into the question, with the curve η_x of the previous calculations, it is seen that from 100° C. values are obtained which are only $\frac{2}{3}$ to $\frac{1}{2}$ of those in the larger chamber. The reason that the value of η is so low is that for certain reasons only a single wheel was employed in the tests, and the ratio $\frac{u}{c}$ was very small. In the comparison only the value of η_x is of influence. In the experiments various types of gas were employed.

If equal heat be supplied per kg. charge, *e.g.* 365 calories per kg., with the smaller chamber

$$t_0 = 100° \text{ C.}$$
$$\eta_x = 26{\cdot}3 \text{ per cent.,}$$

whilst with the larger chamber

$$t_0 = 100° \text{ C.}$$
$$\eta_x = 38{\cdot}3 \text{ per cent.}$$

With small combustion chambers the nozzle opening must be relatively greater than when larger chambers are employed, to obtain equal loss from radiation.

These results lead to the conclusions that gas turbines of small power must be designed with special care, not only as regards the working in the turbine itself, but also with reference to the combustion process.

11. MIXING OF THE EXHAUST GASES AND SCAVENGE AIR.

In the section dealing with scavenge air and the compression of the gas, reference is made to the utilisation of the heat in the exhaust gases as a source of energy for the operation of auxiliary machines.

Consider a regenerator (or economiser) placed between the turbine and the atmosphere, through which the scavenge air and exhaust gases pass, and let

$p_2v_2T_2$ be the condition of the exhaust gases immediately behind the nozzle,
$p_3v_3T_3$ in front of the regenerator,
$p_4v_4T_4$ behind the regenerator,
$p_5v_5T_5$ in front of the exhauster,
$p_6v_6T_6$ behind the exhauster,

then

$$Q_e = \int_{273}^{T_2} c_v dT$$

is the heat remaining in the exhaust gases in the condition $p_2 v_2 T_2$.

It must not be assumed that this heat remains constant during the discharge of all the particles of gas from the nozzle (during an expansion), but rather that

$$Q_e = \int_{G=0}^{G=1} c_v(T_2 - 273)dG = \int_{G=0}^{G=1} c_v t_2 dG,$$

as shown graphically in fig. 58. It must, moreover, be remembered that, when this heat passes out through the last moving wheel, a quantity of heat enters which is sufficient to make up the turbine losses; the exhaust gases, too, at this point possess the kinetic energy corresponding to the exit velocity of the gas from the moving wheel. Expressing this com-

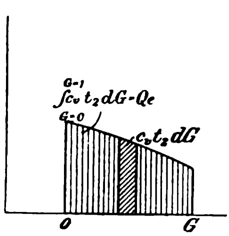


FIG. 58.—$\int c_v t_2 dG$ curve.

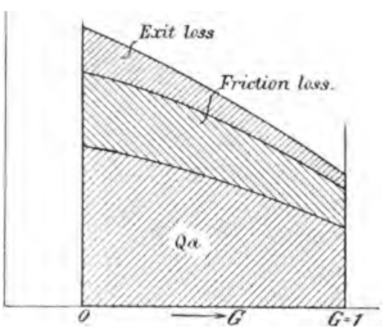


FIG. 59.—Sketch showing heat in exhaust gases.

bined energy in heat equivalent, we have, in the exhaust gases immediately behind the last moving wheel,

$$\Sigma Q_e/_{G=1} = \int_{G=0}^{G=1} c_v t_2 dG + \int_{0}^{G=1} (1 - \eta_{turb}) Q_a' dG.$$

The kinetic energy arising from the last moving wheel does not really appear as heat, as might be deduced from the foregoing equation, but as a form of energy which can, for instance, accomplish the work of compression.

$$\left.\begin{aligned}\int_0^{G=1}(1-\eta_{turb})Q_a'dG=&\int_0^{G=1}(1-\phi_d)Q_a'dG\\ &+\int_0^{G=1}(1-\phi_b)Q_a'dG\end{aligned}\right\}\text{Friction loss.}$$

$$\left.+\int_0^{G=1}(1-\phi_r)Q_a'dG\right\}\text{Loss at exit.}$$

If all these are set out graphically, the form of energy at the exit edge from the last moving wheel for each expansion is obtained (fig. 59). Time diagrams are also given in fig. 60.

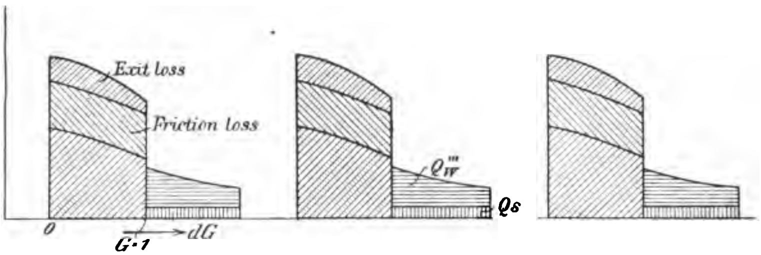


FIG. 60.—Time curve diagrams.

In the scavenging periods the heat supplied by the scavenge air enters. If v kg. of combustion air and s kg. of scavenge air are provided, then

$$G=1+v=\text{Weight of charge.}$$
$$L=v+s=\text{Total weight of air.}$$

$$\frac{s}{1+v}=\text{ratio of weight of scavenge air to weight of charge.}$$

For a charge of 1 kg., the total amount of heat supplied in the scavenge air is

$$Q_s=\frac{s}{1+v}c_v t_s.$$

As this air is taken from the atmosphere, t_s = the temperature of the place where the air is drawn from, $=15°$ C. approximately.

Taking Q_w''' as the heat per kg. abstracted from the walls of the combustion chamber during the scavenge periods, then Q_s+Q_w''' is the total heat content of the scavenge air, per kg. charge. The exhaust gases mix with the scavenge air between the turbine and the regenerator. If Q_w'''' is the heat which is given up during the mixing process by radiation and conduction, the intermittent individual action can be obtained from the resultant continuous process. The heat content of the gas in condition $p_3v_3T_3$ is

$$Q_3=\left(1+\frac{s}{1+v}\right)\int_{273}^{T_3}c_p dT=c_p t_3\left(1+\frac{s}{s+v}\right)$$

and
$$Q_3=Q_e+\text{friction loss}+Q_s+Q_w'''-Q_w''''.$$

Neglecting the last three values, then

$$Q_3=Q_e+\text{friction loss}.$$

$$t_3=\frac{Q_e+\text{friction loss}}{c_p\left(1+\frac{s}{1+v}\right)}$$

$$=\frac{Q_e+\int_0^{G=1}(1+\phi_d)Q_a'dG+\int_0^{G=1}(1-\phi_b)Q_a'dG}{c_p\left(1+\frac{s}{1+v}\right)}.$$

As an example, take the value $g=90$ gr. from Table III., p. 39, for 90 grams of gas per charge; then for 1 kg. charge

$$Q_e=106\cdot6 \text{ calories}$$

and
$$Q_e+Q_f=173\cdot6 \text{ calories per 1 kg. charge.}$$

where Q_f = friction loss.

Therefore
$$t_3=\frac{173\cdot6}{c_p\left(1+\frac{s}{1+v}\right)}$$

and
$$c_p=0\cdot225+0\cdot000058T_3.$$

If $t_3=400°$ C., then $c_p=0\cdot225+0\cdot000058\times673=0\cdot264$

and
$$1+\frac{s}{1+v}=\frac{173\cdot6}{0\cdot264\times400}=1\cdot64;$$

therefore $s=0\cdot64(1+v)$; that is, for 1 kg. charge 0·64 kg. of scavenge air must be provided per charge in order to maintain the temperature of the mixture at $t_3=400°$ C.

Calculating the values for the amounts used per charge, then

Weight of gas	0·090 kg.
Weight of scavenge air	0·253 „
Weight of combustion air	0·305 „
Total weight of air	0·558 „
Weight of gas : Total weight of air . . .	1 : 62

12. AUXILIARY PROCESSES. CIRCULATION OF THE SCAVENGING AND COOLING AIR.

It is evident from the foregoing that the circulation of the scavenging, cooling, and combustion air is an operation of fundamental importance.

Three functions have to be fulfilled: the combustion chamber has to be cleared of the residual gas after combustion, and scavenged with pure air; this includes the cooling of the actual turbine and the combustion chamber, but it must be remembered that the cooling air flows through a separate passage, and not through the nozzles and turbine; this involves some care, owing to its thermodynamic and practical unsuitability; after the completion of this process the pure air remaining in the combustion chamber (this including surplus air) serves for the new charge. In a general form it is not possible to determine how much air is necessary for the actual scavenging and cooling operation, and how much for the charging with air and combustion.

As is shown later, the quantity of scavenge air depends on the final temperature T_2 of the expansion process, on the temperature of the mixed gases maintained in the turbine, the temperature at which the walls are kept, and on the external cooling of the walls. In any case, there is a definite ratio between the quantity of scavenge air and the charge of gas. Similarly, the quantity of the air of combustion (including the surplus) can be expressed in terms of the quantity of gas by known methods. The practical ratio for power gas is given by:

$$\text{Gas : air (total)} = 1 : 4 \text{ to } 1 : 7.$$

The following points may be mentioned as regards the drop of pressure which should be allowed for the air.

The scavenging and cooling air should work with as small a pressure drop as is possible, in order to minimise the power expended; but at the same time this drop must be sufficient to cause the air to pass through the whole turbine. In practice, a drop corresponding to 1–3 metres of water is usually sufficient.

The question of the combustion air stands on a somewhat different basis. The higher the air pressure in the combustion chamber before the admission of the gas, the greater is the relative capacity and power of the chamber under otherwise similar circumstances, and the higher the blast pressure necessary for the gas admission. The greater the specific power of the combustion chamber, the lower is the loss due to radiation of heat compared with the total heat supplied.

A comparison can thus be made by separating the scavenge air from the air of combustion; the scavenge air would have a low pressure drop, whilst that for the air of combustion would be higher. It must, however, be remembered that this is not an actual differentiation, since in reality that part of the scavenge air which remains in the chamber after the completion of the scavenging process, that is, after the closing of the nozzle valve, provides the main supply of the air of combustion; the further quantity of air needed, then enters under a higher pressure. In this comparison, therefore, the further phase should be interpolated of the completion of the combustion air supply by the admission of air at a higher pressure, between the period of the admission of air of combustion at low pressure and the period of charging with gas. On this basis nothing is altered in principle.

Whether this detailed arrangement, however, is thermodynamically of sufficient value to outweigh the undoubted mechanical disadvantages, is a question which can only be solved by an examination of particular cases. It must be investigated by the employment of liquid fuel in the gas turbine.

Relying only on the simple arrangement and operation, with the supply of scavenge, cooling, and combustion air at the same pressure, it would naturally be uneconomical to allow a fall of pressure, greater than is required for the larger portion of the total quantity of air; in most cases, therefore, the pressure drop employed would be 1 to 3 metres water pressure in the gas turbine.

The air is drawn from the atmosphere, and finally discharged into it again. The question then arises as to whether the air should be delivered into the turbine by means of a compressor, so that the air of combustion in the chambers is under a pressure of 1·1 to 1·3 at. abs. (before gas admission), and the pressure p_2 in the turbine is about 1·1 to 1·2 at. abs., or whether the air should be drawn out with the exhaust gases by means of an exhauster, so that the pressures are respectively 1 at. abs. in the combustion chamber and 0·95 to 0·75 at. abs. in the turbine. This question is interconnected with other matters, and no general solution can be given.

From the point of view of actual working, the exhauster arrangement has undoubted advantages, inasmuch as the turbine is then under vacuum; air can be sucked in, but gas cannot leak out through bad packing, etc., and the fact of air entering, for instance, at the end of the turbine where the shaft passes through, is of little importance.

The question, however, to be considered, is the energy required for the air supply, and how this can be most economically provided for. This can be best solved by a utilisation of Ostertag's entropy table for air, whether the auxiliary machine be a reciprocating or turbine blower.

From this table it can be at once determined how much energy is required for the operation of a machine working without loss. The efficiency of a turbine blower (which is almost always used for this work, as a large quantity of air is required at a low pressure) may be taken at 60 per cent. with perfect safety. Recent experiments show that much larger values can be obtained, and as high a figure as 77 per cent. may be reached (Bonte, *Z. d. V. d. I.*, 1910, p. 1661). It is further apparent from the entropy table that the driving energy under otherwise similar conditions is smaller in a machine working without loss, the lower the admission temperature of the air. For instance, if G = 1 kg. in the perfect machine, for the delivery of air from 1 at. abs. to 1·2 at. abs. the energy required at $t = 15°$ C. is

$$AL = c_p(T_1 - T_2) = \cdot 2385 \times 15{\cdot}3 = 3{\cdot}65 \text{ calories,}$$

whilst with $t = 40°$ C., AL = 3·94 calories, or about 7 per cent. more.

The same energy, when working with a compressor, will provide air from 1 to 1·2 at. abs. with the inlet temperature $t = 15°$ C.; if an exhauster be used, it will give a supply of air from ·85 to 1 at. abs. with $t = 100°$ C.; ·88 to 1 at. abs. with $t = 200°$ C.; and ·83 to 1 at. abs. with $t = 15°$ C. The most suitable arrangement for any particular case must be settled in this way, having regard to the fact that the efficiency of the blower increases as the temperature rises. The energy necessary is given from the table for G = 1 kg.-sec., taking 68 per cent. as the efficiency when working adiabatically.

Generally speaking, the turbine should either be arranged for a compressor with air from 1 to 1·2 at. abs., with the admission temperature $t = 15°$ C., or for an exhauster working between ·85 and 1·0 at. abs., with $t = 100°$ C., or their equivalent. The expenditure of energy for the compressor or exhauster with a ratio of gas : air of 1 : 4 is 122·2 H.P., and with the ratio 1 : 7, 214 H.P., whilst the output of the gas turbine * whose over-all efficiency is 30 per cent., and in which G = 1 kg. per sec. 1200 calories per sec., is

$$\frac{1200 \times 0{\cdot}30 \times 3600}{637} = 2040 \text{ eff. H.P.}$$

In other words, the expenditure of energy for the supply of air with a ratio of gas to air of 1 : 4 is 5·96 per cent. of the output of the gas turbine, and when the ratio is 1 : 7 the percentage is 10·5 per cent.

These relative values are low if the combustion chamber, or that portion of it actually enclosing the combustion space, is very effectively cooled, and the mixed gases are maintained in the actual turbine at a relatively high temperature. The more the combustion chamber walls

* This assumption is not quite correct, as the efficiency is rather higher when an exhauster is used than with a compressor.

are cooled by the air, and the lower the temperature of the mixed gases in the turbine, the higher are the relative values given above.

The importance of these questions is, however, diminished from the point of view of actual operation, if the heat in the exhaust gases is utilised for the production of energy for the auxiliary machines. This regenerative process is discussed later.

AUXILIARY PROCESSES.

13. COMPRESSION OF THE GAS AND GAS MIXTURE.

The questions which enter into consideration in dealing with the other subsidiary matters, such as compression of the gas and gas mixture, are quite different from those already examined.

Nothing is altered in principle, if in order to render comparisons simpler the pure gas is compressed alone, and the pressure of the gas mixture before ignition is raised by the pre-compressed gas entering the closed combustion chamber, mixing with the air in it, and at the same time raising the pressure of the gas mixture; or if the compressed gas and air enter the chamber together. In either case the condition $p_0 v_0 T_0$ is reached immediately before ignition, so that a good mixture of gas with the air in the chamber is obtained.

As, however, the compression and delivery of an explosive mixture, even with most careful construction, are sources of danger in operation, which are completely avoided by previous compression of the pure gas, only such compressions of gases free from oxygen will be considered in the following discussions. Unfortunately, no entropy tables are available which would be helpful to a clear examination of these questions. The entropy tables of Ostertag serve only for air, and those of Stodola are mainly for relatively high temperatures. The questions must therefore be explained by analytical methods.

Generally speaking, the gas pressure in the reservoir from which the gas flows to the individual combustion chambers must be maintained at such a figure that the pressure p_0 and the temperature T_0 are reached in the time available for the admission of the charge, and the necessary quantity of gas blown in. The pressure p_g must therefore be equal to or greater than p_0. The best ratio of $p_g : p_0$ cannot definitely be given, as in every special case the gas valve opening, time of admission, and completeness of the mixing of the gases have an effect. In general, it is satisfactory to take

$$p_g = p_0 + 0{\cdot}2 \text{ at. abs.}$$

Nothing has been mentioned regarding p_0. The current opinion is that p_0 should be as high as possible, as in the reciprocating engine a pressure of 12 atmospheres is allowed, and in the Diesel motor 30 atmospheres; this is not possible with the gas turbine, which therefore might seem to be at a disadvantage.

The value of this opinion is minimised by the fact that the reason for making p_0 high in reciprocating engines is not clearly understood. It is possible in the reciprocating engine, with its limited compression ratio, to reach the efficiency actually attained by pre-compression alone; it could also be obtained by working with a lower bottom pressure, which would, however, necessitate very long cylinders. Moreover, the specific capacity could be increased by raising p_0 just as in the gas turbine.

It is to be noted, on the other hand, that heat is transmitted to the cylinder walls during combustion and expansion. Bearing this in mind, we may compare two motors, one designed for a high pre-compression and the other for a lower one; both cylinders work with the same charge, the compression ratio is the same, and also the speed of revolution.

As the maximum temperature obtained during the combustion, apart from the heat supply, depends only on the temperature before ignition, this maximum temperature will be the same in both cases, assuming the pre-compression to be isothermal. The final temperature of expansion will also be identical in both systems, as the compression ratio is the same, in the perfect machine.

The heat transmitted to the walls will thus depend only on the inner surfaces of the two cylinders, and from this point of view the cylinder arranged for high pre-compression is far superior to the other.

This is the real basis of the success of the compound arrangement with reciprocating internal combustion engines, and it is from purely constructional considerations that reciprocating engines are designed for high pre-compression.

The thermodynamic advantages of an increased compression ratio can be obtained in a gas turbine in a perfectly natural way by increasing the bottom pressure range. This may be accomplished very easily and simply.

In the piston engine p_2 is about 6 to 8 at. abs., whilst in a gas turbine provided with an exhauster the value is about 0·85 at. abs. Thus, in the gas turbine, as was seen from the results of the experiments, an efficiency can be obtained equal to or greater than that of the reciprocating engine, with a much lower value of p_0. It does not follow, of course, that the gas turbine cannot operate with a higher value of p_0, or can only derive an advantage by increasing p_0, which would be neutralised by the increased expenditure of energy in work of compression. The efficiency of the turbine is naturally raised with a higher value of p_0. The limit of p_0 is chosen in individual cases such that the work of compression required can just be carried out from the complete utilisation of the heat in the exhaust gases from the turbine. This always gives the highest efficiency.

For power gas whose volumetric composition is 5·8 per cent. CO_2, 15·2 per cent. H_2, and 25 per cent. CO, the following values may be taken:—

$$\left.\begin{array}{l} c_v = 0{\cdot}198 \\ c_p = 0{\cdot}278 \end{array}\right\} \begin{array}{l} \gamma = 1{\cdot}41 \\ t = 30^\circ \text{ C. (Langen's figure).} \end{array}$$

The energy required for the compression of the gas if $t = 30^\circ$ C., with $G_{gas} = 1$ kg. per sec., is therefore

$$AL = {\cdot}278\,(T_a - T_e)$$

and

$$\frac{T_a}{T_e} = \left(\frac{p_g}{1}\right)^{\frac{\gamma-1}{\gamma}}.$$

Taking $p_g = 1{\cdot}6 \quad 1{\cdot}8 \quad 2{\cdot}0 \quad 2{\cdot}2$ at. abs.,

then $\frac{T_a}{T_e} = 1{\cdot}144 \quad 1{\cdot}183 \quad 1{\cdot}219 \quad 1{\cdot}285$;

since $T_e = 303$, therefore

$$\begin{array}{rllll} T_a = & 348 & 360 & 369 & 380 \\ T_a - T_e = & 45 & 57 & 66 & 77 \\ AL = & 12{\cdot}54 & 15{\cdot}85 & 18{\cdot}4 & 21{\cdot}4. \end{array}$$

These values are for single-stage adiabatic compression in a machine working without loss.

Calculating from the efficiency with adiabatic compression, $\eta_a = 68$ per cent. (for turbine blower with intercooling), the energy required for 1 kg. of gas per sec. is respectively 104·5 H.P., 132·4 H.P., 154 H.P., 179 H.P.

The total driving power for the gas blower and the air circulator may be taken in practice as 12–15 per cent. of the output of the gas turbine.

It should here be added that the determination of the supply of energy to the auxiliary machines depends on how much heat is given by the regeneration process of the gas turbine. If this point is neglected, the power supplied to the auxiliaries should naturally be as small as possible.

The question may be raised as to the most suitable type of blower to employ for the compression of the gas. Turbine blowers are only adapted for the supply of a relatively large quantity of gas or air. Not only is the cost high, but for capacities of less than 3000 cubic metres per hour turbine blowers are uneconomical. In such cases only piston blowers need be considered, as they give a higher efficiency than the turbine blowers. According to the output of the gas turbine and the type of gas employed—power gas, blast-furnace gas, or coke-oven gas of high calorific power—various types of blowing plant may be used, such as steam turbine and turbo-exhauster; gas turbo-compressor or steam turbine; turbo-exhauster and steam engine; reciprocating gas blower or steam engine; exhauster; reciprocating gas blower. Similar variations are usual for condensing plant in steam-power installations. The purity of the gas also has an influence on the choice of the gas blower. If it contains tar, it cannot be employed with a turbo-blower, as this would become foul very rapidly, and cleaning is comparatively difficult. If liquid fuel is used, a fluid pump must be employed instead of a gas blower, as an auxiliary to the compressor for the injection of the air. The power required for this pump diminishes, contrary to that for the compressor for the air injection; but this power is in no case less than that for the equivalent gas blower for power gas.

Taking, for instance, from the example given above the value 132 H.P. for $p = 1{\cdot}8$ at. abs. and 1 kg. of gas per sec., the equivalent is 0·12 kg. of oil per sec. With the same driving power the injection air compressor can be operated, and provides an injection and atomising pressure of about 6 atmospheres. Experiments show that this pressure is sufficient to atomise efficiently heavy oil with the given ratios in the gas turbine, but it is not too high.

The same power is therefore required for the auxiliaries when using liquid fuel as with power gas, if the atomising is accomplished by means of air to bring the liquid fuel in a suitable condition for ignition.

14. UTILISATION OF THE EXHAUST HEAT.

After the exhaust gases mix with the scavenge air, they pass continuously through the regenerator or economiser to the exhaust, and thence to the atmosphere, or direct into the atmosphere if a compressor be employed. In the regenerator as much heat as possible should be extracted from the mixed exhaust gases and scavenge air. This regenerated heat should then be utilised for operating the auxiliary machines by the aid of water and steam. The transmission of the heat to water can be accomplished by direct mixing, or the water can be circulated in pipes and the exhaust gases and scavenge air pass round the pipes. In the first case the regenerated heat is available in water and steam naturally at a pressure of less than p_2, whereas in the second case the steam pressure can be much higher. As, for a good efficiency in the gas turbine, p_2 should be as small as possible, the case is only examined later in which the gases and the water are kept separate.

Whilst the exhaust gases and scavenge air in the regenerator change from the condition $p_3v_3T_3$ before reaching the regenerator to $p_4v_4T_4$ behind the regenerator, let the water enter the pipes at, say, 25° C.; its temperature is raised to 100° C., and steam is generated at a pressure of, say, 9 at. abs., the temperature being 174·4° C. (fig. 61).

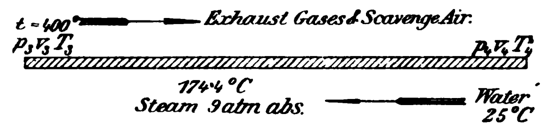


FIG. 61.—Sketch showing changes in regenerator.

Considering only the process through which the exhaust gases and scavenge air pass, the variation of heat content is obtained generally from the following equation:—

Heat content before the regenerator = heat content behind the regenerator + heat in the water + compression or expansion work;

or in general from

$$dQ = di - AvdP.$$

If p is constant or $p_3 = p_4$,

then $$dQ = di = c_p dt,$$

and the heat from the water (steam)

$$Q_r = \int_{T_4}^{T_3} c_p dT = c_p t_3 - c_p t_4.$$

Considering now the process passed through by the water and steam;

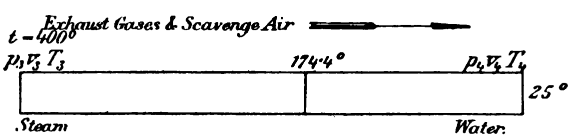


FIG. 62.—Sketch showing heat passage in regenerator.

483·1 calories are required to evaporate 1 kg. of water from 174·4° C. at 9 at. abs., and to raise the water from 25° to 174·4° the heat required is

$$176{\cdot}6 - 25 = 151{\cdot}6 \text{ calories},$$

or a total of 634·7 calories per kg. of steam. If d kg. of steam be raised per 1 kg. of exhaust gases, then

$$\left(1 + \frac{s}{1+v}\right)\left(c_p t_3 - c_p t_4\right) = 634{\cdot}7d.$$

Dividing up the operations, it may be stated that the exhaust gases + the scavenge air can only raise steam by transmission of heat so long as the temperature of the exhaust gases + scavenge air is equal to or greater than 174·4° C.

It follows that

$$\left(1+\frac{s}{1+v}\right)\left(c_p t_3 - c_p \times 174{\cdot}4\right) = 483{\cdot}1 \times d$$

and

$$\left(1+\frac{s}{1+v}\right)\left(c_p \times 174{\cdot}4 - c_p t_4\right) = 151{\cdot}6 \times d,$$

Values previously given may be inserted here: *

$$1+v = 1 \text{ kg.}$$

$$s = {\cdot}64 \text{ kg.}$$

$$c_p \times 174{\cdot}4 = 174{\cdot}4({\cdot}225 + {\cdot}000058 \times 174{\cdot}4) = 41{\cdot}1$$

$$c_p t_4 = 41{\cdot}1 - \frac{151{\cdot}6d}{1{\cdot}64}.$$

Further, during evaporation

$$1{\cdot}64(0{\cdot}264 \times 400 - 0{\cdot}235 \times 174{\cdot}4) = 483{\cdot}1d$$

$$d = \frac{1{\cdot}64 \times 64{\cdot}5}{483{\cdot}1} = 0{\cdot}219.$$

With 1 kg. exhaust gases and ·64 kg. scavenge air, ·219 kg. of steam can therefore be generated at 9 at. abs. pressure. The temperature t_4 is determined from

$$c_p t_4 = 41{\cdot}1 - \frac{151{\cdot}6 \times 0{\cdot}219}{1{\cdot}64} = 20{\cdot}9$$

$$({\cdot}225 + {\cdot}000058T_4)(T_4 - 273) = 20{\cdot}9$$

$$t_4 = 97° \text{ C. } \dagger$$

For this case, therefore, with ·090 kg. of gas per charge
and ·558 kg. of air (total) per charge,
·0528 kg. of steam at 9 at. abs. can be generated per charge.

Calculating these values for 1 kg. of gas, the corresponding figures are:

1 kg. of gas,
6·2 kg. of air (total),
·59 kg. of steam at 9 atmos. abs.

If the steam thus generated be delivered to a steam turbine driving

* Page 51.

† This agrees with experimental results.

the auxiliary machines, and if η is the over-all efficiency of the auxiliary plant (exhauster, compressor, steam turbine), then

$$\eta J \times 0{\cdot}59$$

is the regenerated heat in the form of compressed gas or exhausted gas mixture.

If the steam turbine works on a vacuum of 90 per cent., $J=162$ calories; and if $\eta=40$ per cent., the over-all efficiency of the gas blowing plant, then $\cdot 59 \times 0{\cdot}4 \times 162 = 38{\cdot}3$ calories can be regenerated in the form of compressed gas or exhausted gas mixture.

If the exhauster has to suck air at ·89 at. abs. and 97° C. into the atmosphere, the expenditure of energy necessary per kg. is 3·1 calories, and the total energy for the supply of all the air is

$$3{\cdot}1 \times 6{\cdot}2 = 19{\cdot}22 \text{ calories.}$$

For the compression of the gas there remains

$$38{\cdot}3 - 19{\cdot}22 = 19{\cdot}08 \text{ calories.}$$

With this amount the gas can be compressed to about $p_g = 2{\cdot}0$ at. abs. with the employment of a turbo-blower, and $p=$ about 3 at. abs. if a reciprocating blower be utilised. This example is based on a value of $p_0 = 1{\cdot}67$ at. abs.

It can be seen from the foregoing concrete example that the requisite energy for the operation of the exhauster and the gas blower can be conveniently obtained by means of the regeneration process.

It may thus be taken that the efficiency of the gas turbine calculated above is not prejudicially influenced by the auxiliary plant — the exhauster and the gas blower. The necessary energy for the operation of these auxiliaries can be completely provided by means of the regeneration process. The fundamental importance of this regenerative operation is thus clearly apparent.

It can be shown that the necessary pipe surface of the regenerator is within practical limits. For the transmission of the heat of the exhaust gases and scavenge air to the water the following equation may be used:

$$Q_r = C_1 F z\left[\left(\frac{E}{100}\right)^4 - \left(\frac{T}{100}\right)^4\right]$$

and

$$Q_c = aFz(E - T),$$

in which $C_1 = 4$,

$F=$ surface of pipe coil in sq. metres.
$z=$ time in hours,
$E=$ absolute temperature of gas,
$T=$ absolute temperature of water,
$a = 2 + 10\sqrt{v}$.

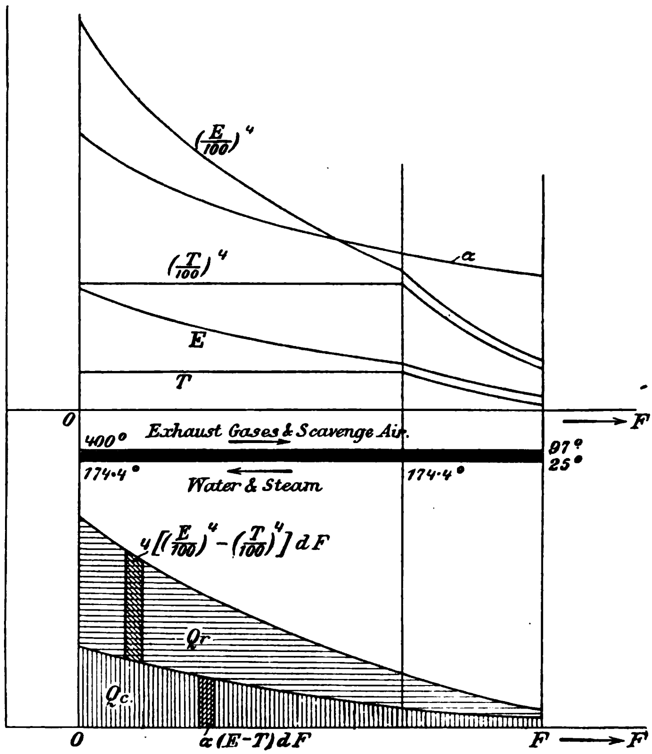
The heat transmitted per hour is therefore

$$Q = \alpha F(E-T) + 4F\left[\left(\frac{E}{100}\right)^4 - \left(\frac{T}{100}\right)^4\right].$$

Strictly speaking, the process must be followed step by step from the following curve :—

$$dQ = \alpha(E-T)dF + 4\left[\left(\frac{E}{100}\right)^4 - \left(\frac{T}{100}\right)^4\right]dF = c_p dE.$$

Taking G gas = 1 kg. per sec. in the foregoing example, and calculat-



FIGS. 63 and 64.—Curves showing action of regenerator.

ing the corresponding values for the regenerator, the following is approximately true :—

$$\alpha = 2 + 10\sqrt{v} = 2 + 10\sqrt{27{\cdot}8} = 55 \text{ approximately},$$
$$E_m = E_{mean} = \sqrt{E_a E_b} = 407,$$

where E_a = initial absolute temperature of gas,

E_b = final „ „ „

or $\frac{E_m}{100} = 4{\cdot}07.$

$$T_m = T_{mean} = \sqrt{T_a T_b} = 365,$$

where T_a = initial absolute temperature of water,
T_b = final " " "

$$E_m - T_m = 42$$

$$\left(\frac{E_m}{100}\right)^4 - \left(\frac{T_m}{100}\right)^4 = 275 - 178 = 97$$

$$Q_r'' = 89{\cdot}4 \times 3600 = 55 \times 42 \times F_r'' + 4 \times 97 + F_r''$$

$$F_r'' = 119{\cdot}5 \text{ sq. metres.}$$

At the beginning of the passage of heat the following holds good :—

$$\alpha = 63 \text{ approximately}$$
$$E_m = E_{mean} = \sqrt{637 \times 447} = 548$$
$$T_m = T_{mean} = 447$$
$$E_m - T_m = 101$$

$$\left(\frac{E_m}{100}\right)^4 = 912\ ; \left(\frac{T_m}{100}\right)^4 = 404$$

$$\left(\frac{E_m}{100}\right)^4 - \left(\frac{T_m}{100}\right)^4 = 508$$

$$Q_{1r} = 285 \times 3600 = 63 \times 101 \times F_r^1 + 4 \times 508 \times F^1$$
$$F_r^1 = 122 \text{ sq. metres.}$$

$F_1{}^r + F_{11}{}^r = F = 241{\cdot}5$ sq. metres, which is the pipe area required for the regenerator coil for a gas turbine with an output of about 2000 effective H.P., or about 0·12 sq. metre regenerator area per effective H.P.

In surface condensers for steam turbines the cooling surface is taken on an average as about 0·11 to 0·12 sq. metre per effective H.P., or approximately the same ratio as for a gas turbine.

It follows, therefore, that the pipe area required for the regenerator is within practical limits, and that from this point of view there is nothing to be said against the adoption of a regenerator.

As long as the temperature of the mixed scavenge air and exhaust gases is not below 100° C., there is no danger that the sulphurous acid in the exhaust gases will be in liquid form and cause damage to the metal. This has never occurred with the first plant. From this standpoint it is not exactly advantageous to allow the gases to mix directly with water.

15. PRINCIPLES OF REGULATION.

As has been clearly shown in the foregoing pages, for maximum over-all efficiency in a gas turbine, the combustion chambers must come into operation in definite and regular sequential periods. The efficiency of combustion has a maximum value for a certain charge of gas, and the nozzles and blading must be arranged to suit the corresponding values of the pressure or heat drop. As the efficiency decreases on both sides of this definite pressure or heat drop, it is apparent that the curve of over-all efficiency falls even more rapidly on both sides of the most favourable pressure or heat drop.

Generally, it may be taken as a principle of the gas turbine that the pressure or heat drop shall be maintained as near as possible to the most favourable value.

This can only be accomplished if a number of combustion chambers be arranged in a circle, and the regulation is then provided for by cutting out one chamber after another.

Admission to the turbine must thus be always more irregular, the more chambers there are put out of action. If it be necessary for the turbine to run with a certain degree of regularity even on no load, a certain minimum number of chambers must be in operation when running light. If the regularity is, for instance,

$$\frac{\Delta w}{w} \leqq \frac{1}{600},$$

it follows that, if L is the energy supplied to the turbine wheel in metre-kg. per charge, and E the moment of inertia of the turbo-dynamo,

$$\frac{\Delta w}{w} = \frac{\Delta L}{E w^2} \leqq \frac{1}{600}.$$

From this equation ΔL can be calculated for any turbine. For example, in a 1000-H.P. machine at 3000 r.p.m., if $E = 8$, then $E w^2 =$ 790,000 metre-kg., and therefore for $\frac{\Delta w}{w} \leqq \frac{1}{600}$

$$\Delta \leqq 1320 \text{ metre-kg.}$$

If the areas L be drawn on a time diagram, these must follow each other sufficiently rapidly that the difference compared with a constant energy supply does not exceed this value ΔL.

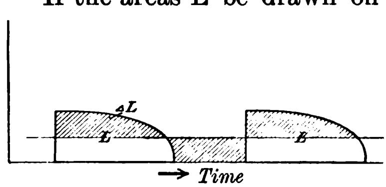


FIG. 65.—Curves of irregularity.

From this, the periods can be determined which are permissible between two consecutive explosions.

If the power which can be obtained from the chambers is larger than is required when the turbine is running light, the quantity of gas per charge must be reduced.

The regulating process thus described requires a "hit-and-miss" governor in combination with a gas-throttling regulation.

The "hit-and-miss" governing alters the number of combustion chambers coming into operation in a given time, whilst the throttling regulation varies the pressure of the gas before the charge enters the individual chambers. Another method of governing is one in which the charge per second for every separate combustion chamber remains unaltered in point of time, but the periods between the charging of the chambers one after the other, are regulated; when the maximum period is reached which will allow of uniform running, the throttle governor comes into action.

This latter method in combination with throttle governing has the advantage over the "hit-and-miss" arrangement in that all the chambers are in operation, and the heat is supplied regularly. If the "hit-and-miss" type of governor be employed working with a throttle governor, care must be taken in the construction to avoid any trouble through expansion of the portions of the turbine in operation. This is, generally speaking, unnecessary with the internal method of regulation, combined with a throttle governor.

Both methods, however, comply with the conditions previously laid down for the regulation of a gas turbine: the pressure or heat drop is in both cases very near the most favourable value. A less suitable method is that in which the intervals between the individual admissions are regulated, and also the actual times of admission, combined as before with gas-throttling regulation. Governing purely by throttling of the gas is simple but inefficient.

The blowing plant must also be regulated with the gas turbine. Since the energy for driving this plant is obtained from the exhaust heat, it is advantageous that the blowing plant be governed by maintaining the steam pressure constant in the regenerator. If, for instance, the gas turbine is working with a big load, more driving energy will be given to the blowing plant; the speed of revolution of the blower rises, and with it the compression pressure or exhauster pressure, and *vice versa*.

16. SUMMARY.

INFLUENCE OF THE INDIVIDUAL FACTORS ON THE PROCESS.

The foregoing theory of the gas turbine does not serve as a guide for the calculations of the turbine, but explains the influence of the various factors on the gas turbine process.

In this theory, after a short description of the cycle, the individual phases of the process are analytically examined, such as ignition, combustion, and expansion. It is shown that the general treatment, without any consideration of the heat given to the walls, leads to entirely erroneous results; the theory further shows that this factor is of fundamental importance to the gas turbine, and that it varies according as the gas turbine cycle is a mechanical or heat cycle. The employment of the entropy diagram for these phases shows how extremely clearly, simply, and concisely the results of the gas turbine cycle can be expressed.

Later, the actual working cycle in the turbine is examined, this being naturally analogous to the steam turbine cycle, particularly to the early type of Parsons turbine, working with intermittent steam blasts. It is made specially clear that the actual operation must be compared with the working of a machine running without loss in which the available drop decreases with an increasing weight of the discharged gases, and not one in which the available drop remains constant. Included in this is the general discussion of the scavenging process. It is shown that the combustion chamber must be scavenged between two consecutive admissions, in order to have a low temperature, and a pure mixture before ignition; it is demonstrated that this is a fundamental condition for the gas turbine process, which, if not fulfilled, leads to the cycle being a heat cycle and not a mechanical one. From this point of view the turbine working with continuous combustion is favoured.

The discussion on the process of charging (or admission of gas) shows the very complicated operations which are involved during the admission of gas and air mixture to the combustion chamber. The absolute necessity of closing the combustion chamber at its turbine outlet during admission is clearly demonstrated.

In the section on the " Explosion Wave," the discussion of the question of the detrimental sudden explosion is based on the calculations used in physical chemistry. It is shown, from experiments with carbon monoxide and a mixture of power gas and air, how the curves for the combustion pressure and the sudden explosion pressure, depend on the temperature before ignition; how, above a certain temperature, the critical compression curve lies below the combustion pressure curve to a greater and greater extent, and hence a larger portion of the gas is exploded suddenly. A series of gas turbine time-pressure diagrams are given, showing the explosion and combustion pressures, following the nomenclature adopted in physical chemistry.

In the next section the influence of the temperature before ignition is shown by calculation from two examples, these examples being taken from tests on gas turbines. The individual entropy diagrams and the efficiency curves are given, and it is shown that the best results are obtained when the temperature before ignition is as near as possible to the outside temperature of the air.

The influence of the gas charge is then demonstrated by means of calculations from two examples, which are also derived from experiments on gas turbines. It is seen that an efficiency of 30 per cent. can be realised in the gas turbine, calculated from the gas and the effective H.P.-hours. In this case also the relative values are demonstrated by means of entropy diagrams and efficiency curves. The dependence of the efficiency on the temperature before ignition, and on the charge of gas, is illustrated in a diagram drawn with three sets of co-ordinates.

In the next section the influence of the size of combustion chamber is examined, also by means of an example taken from practical tests (with a relatively small gas turbine), lighting gas being employed. In the calculation all the data and the various steps in the analysis are given, so that it is quite clear how all the examples are worked out. The calculations were in agreement with the practical results, and showed that the efficiency of combustion is considerably less with a smaller chamber than with a larger one, other conditions being equal. It is further made clear that gas turbines for small powers are much more difficult, and need greater care in construction, not only because of the actual working cycle in the turbine, but also on account of the combustion arrangements.

The circulation and delivery of the scavenge and cooling air are discussed in the following section. It is demonstrated that the pressure drop for the scavenge air should be as small as possible, and only sufficient to allow it to pass through the turbine. The methods of operation—with a compressor or exhauster—are compared, and a few examples worked out.

In the next section compression of the gas and mixture is discussed in a similar manner. The question is raised as to the reason that a high efficiency in a reciprocating engine necessitates a high compression, and why this is not the case in a gas turbine; it is explained that the reasons for this do not lie in the principle of the cycle, but are based on constructional considerations.

A few examples follow, and an examination of them shows that a gas turbine working on liquid fuel requires about the same energy for the operation of its auxiliary machines as when running on gas, so long as the liquid fuel is injected and atomised by means of air, which experiments showed to be by far the most suitable for a gas turbine.

In the section on the mixing of the exhaust gases with the scavenge air, the action after leaving the turbine is discussed, and the temperature of the mixed gases calculated for the general case and for particular examples. It is shown that a temperature of the mixed gases may be employed which will give a very satisfactory operation without too large a quantity of scavenge air being needed, this being also proved by experiment.

This temperature of the mixed gases in the exhaust pipe of the turbine must not be confused with the temperature at the end of the expansion.

In the last section but one of the theory of the gas turbine, the question of the regeneration of the exhaust heat is treated. This is done by general analytical methods, and by examples it is shown that the auxiliary machine for the gas turbine can be driven without any trouble from the energy in the exhaust, so that the efficiency of the turbine as calculated is not in the least diminished by the expenditure of energy for the auxiliary machines. In general, the driving energy for these auxiliary machines should be made dependent on the regenerated exhaust heat. It is also shown that the regenerator required for this work can be of dimensions not greater than those of a surface condenser for a steam turbine of equal size.

In the final section the regulation of the gas turbine is examined. It is shown that the method of regulation of the intervals between consecutive admissions, combined with throttle governing or a "hit-and-miss" regulation also with throttle governing, is the most suitable for a gas turbine; the pressure or heat drop for each process being maintained at the most suitable value for the turbine.

PART II.

CONSTRUCTION OF THE GAS TURBINE.

1. GENERAL ARRANGEMENT.

IN Part I. of this volume, in the "Theory of the Gas Turbine," the conditions are clearly demonstrated which are essential in order to render a gas turbine an efficient prime mover—efficient both in its utilisation of the energy supplied and also as regards reliability of operation.

The construction of the gas turbine will now be discussed, and naturally the principles can be carried out with different methods of construction. It is, therefore, not implied that these fundamental conditions for the gas turbine cycle can only be complied with by following on the lines of the constructional details described below. It may be stated, however, that among the various possible methods of construction the following is on the whole the most suitable.

The gas turbine may be built as a vertical or horizontal machine, and there are no essential reasons to show that the one design has advantages over the other. The steam turbine, for instance, is constructed in Europe with a horizontal shaft, and in the United States with a vertical shaft (except for small units in certain cases). The question may therefore be said to depend on personal inclination and prevailing opinion. For the original gas turbine and the first turbine for dynamo driving, the vertical construction was adopted (figs. 66 and 67).

Above the ring of combustion chambers the support for the dynamo is arranged with a sleeve-bearing for the turbine shaft; on the support, the dynamo is fixed, and on this the top bearing block with a sleeve-bearing and the footstep. The turbine shaft is thus supported freely from above, and the rotor is floating. On the turbine casing no other constructional element is supported. A neck gland is unnecessary, as the turbine is run in combination with an exhauster.

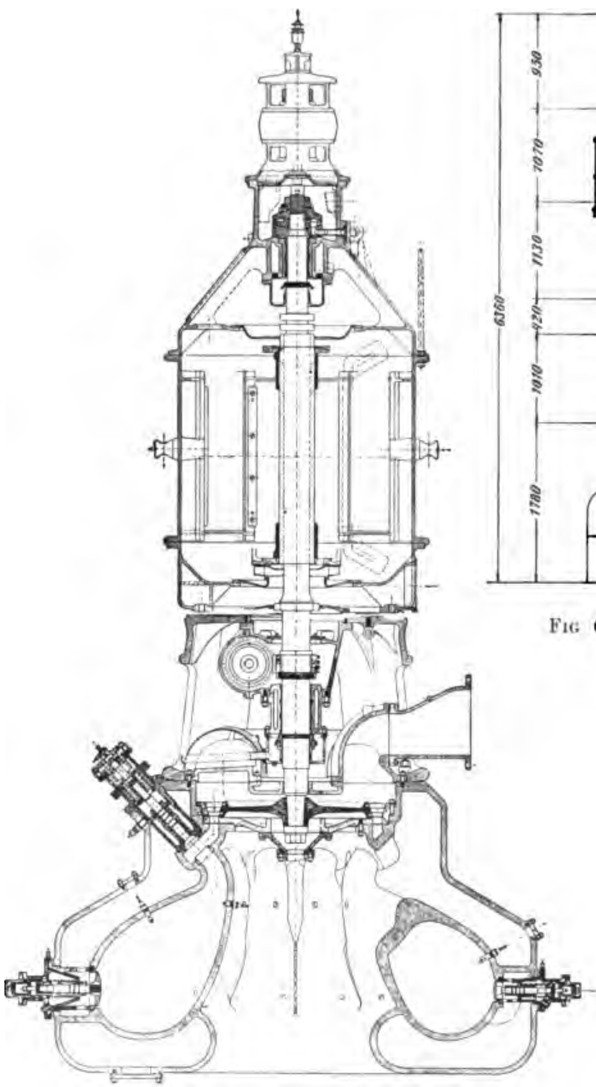


Fig. 66.—Section through a gas turbine of 1000 H.P., with dynamo.

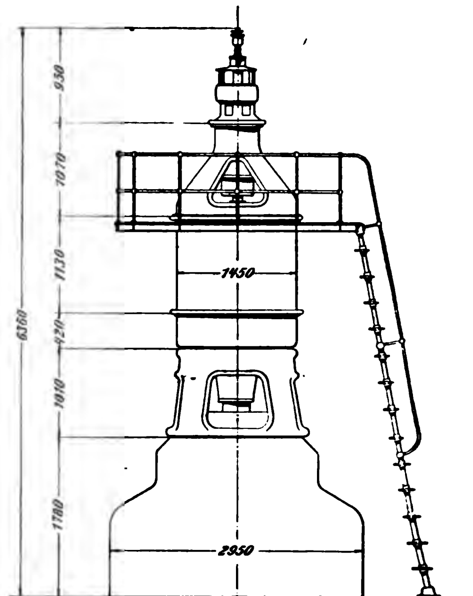


Fig. 67.—External view of 1000-H.P. gas turbine and dynamo.

The turbine rotor and the rotor of the dynamo are fixed on a through shaft without any coupling.

A bearing is arranged for the rotor, supported on the centre of the combustion chamber ring, and ensures equal expansion of the

FIG. 68.—Rotor of 1000-H.P. machine.

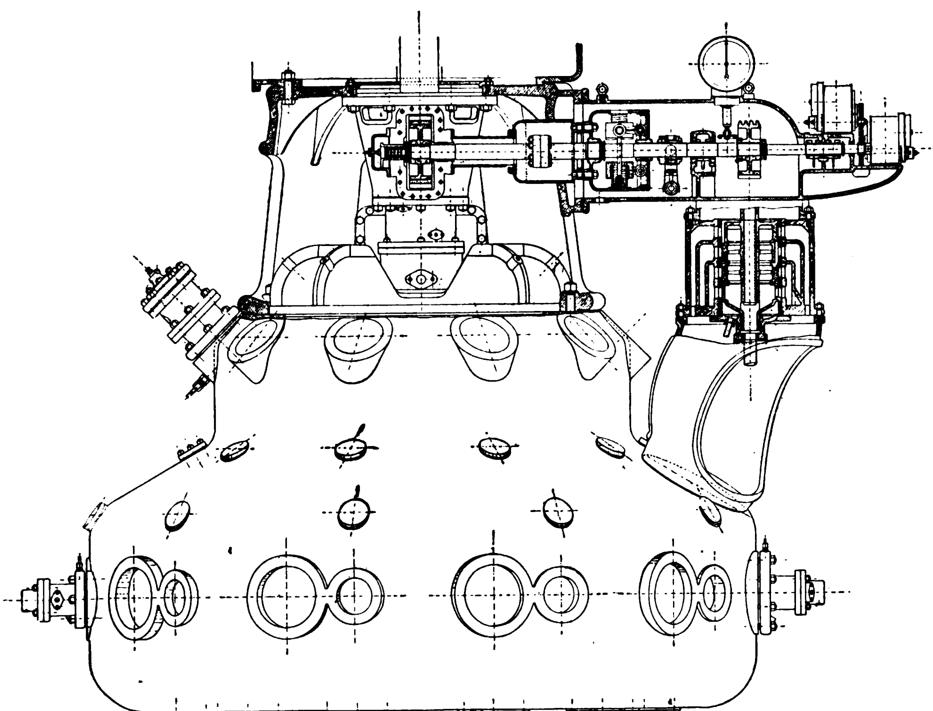


FIG. 69.—Section through governor shaft of 1000-H.P. turbine.

rotor and the bearing, so that there is no vibration when the turbine is running.

All the valves are situated in the combustion chamber ring, and are individually removable without any further dismantling being necessary (fig. 69).

The horizontal governor shaft is contained in a casing bolted to the dynamo support. It is driven through a pair of helical wheels between the bottom journal and the dynamo.

Fixed on the governor shaft are the main and safety governors (which latter cuts off the ignition in a very simple manner), the

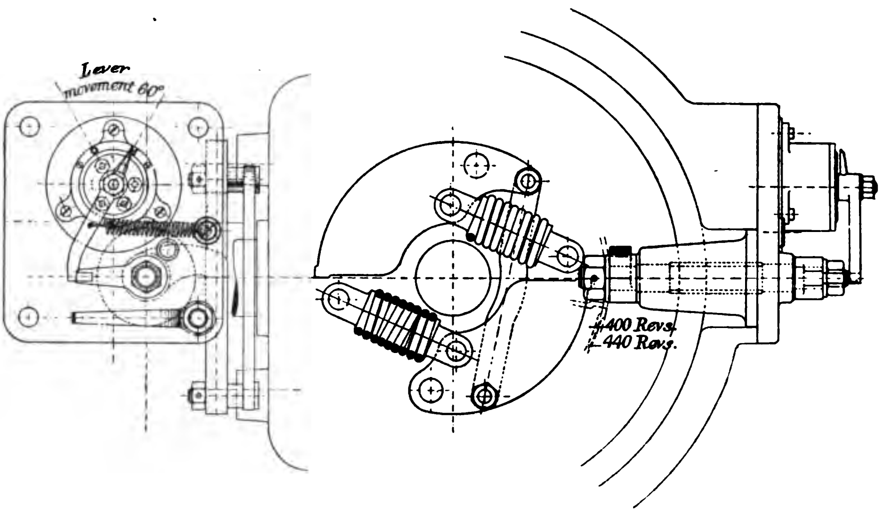


FIG. 70.—Safety governor.

tachometer, the spur-wheel drive for the vertical oil-distributor, and the drive for the ignition arrangements.

The oil-distributor for the rotor and stator is supported from the combustion chamber ring. The gas-throttling valve with the arrangement for the "hit-and-miss" regulation is also bolted to the combustion chamber ring.

The whole machine is thus self-contained, without necessitating a bedplate for the various portions.

Variations may be made with the vertical design by arranging for the oil-distributor to be driven, for instance, off the blower shaft instead of direct from the gas turbine. If the governor is then arranged on the top bearing, no projection is necessary on the combustion chamber ring.

This does not involve any disadvantages, as there is no connection between the turbine shaft and the valve gear, unlike a reciprocating engine.

In a horizontal gas turbine the air and gas valves would be arranged on the side of the combustion chamber ring, and not around the peri-

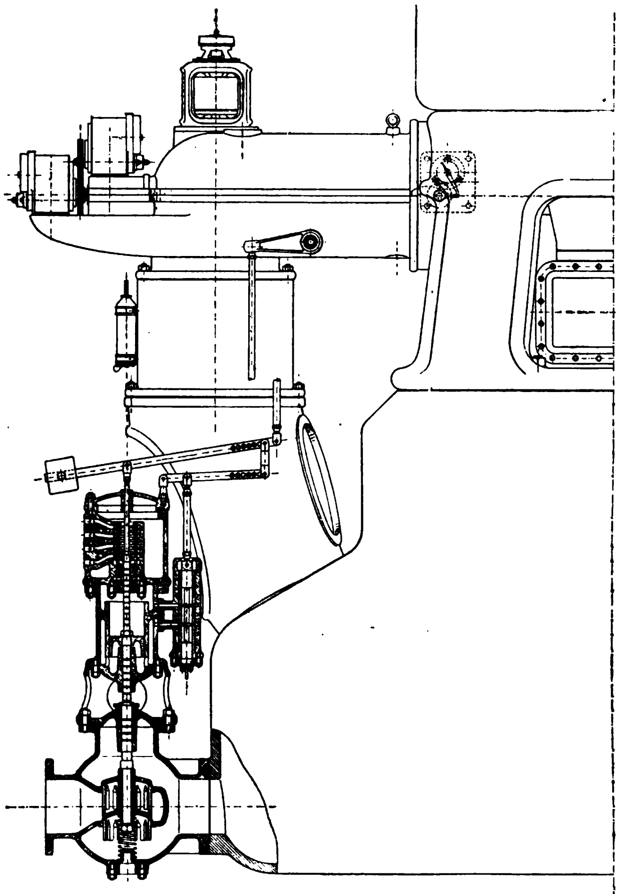


FIG. 71.—Section through governing mechanism attached to the gas turbine.

phery as in a vertical machine. The nozzle valves (flap valves) are built in the horizontally divided turbine casing. The rotor is supported freely and is accessible by removing the top portions of the turbine casing. The shaft is in one piece with the dynamo shaft as before, and supported in two bearings, one on each side of the dynamo. The governor is arranged between the dynamo and turbine.

With this arrangement the running is also free from vibration, as no portion of the shaft can expand relatively to the bearings. Where the shaft passes through the turbine casing a large amount of play may be allowed if an exhauster is employed, as no neck gland is necessary.

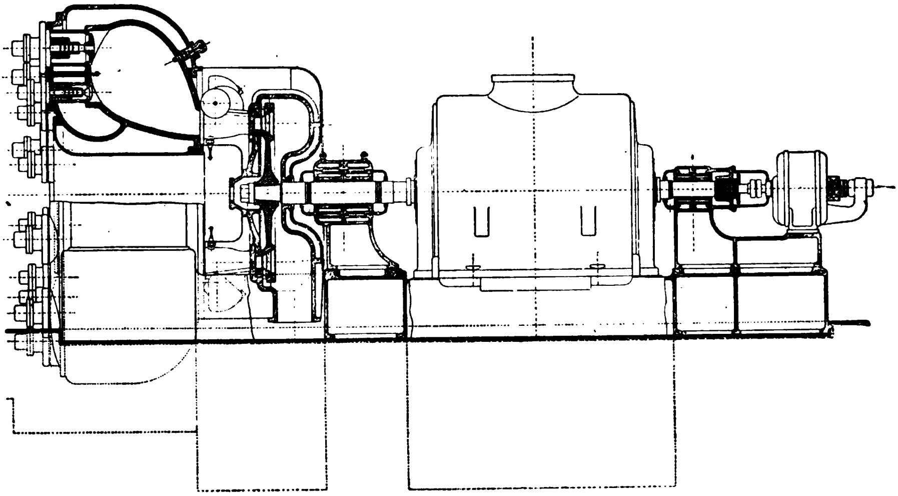


FIG. 72.—Section through 1000-H.P. horizontal gas turbine, with dynamo.

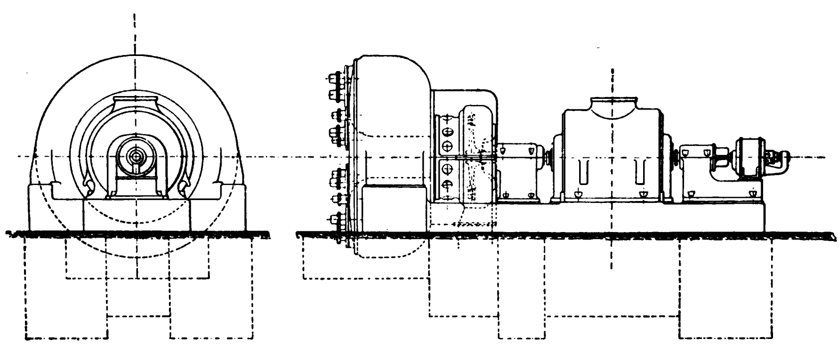


FIG. 73.—1000-H.P. horizontal gas turbine and dynamo. General arrangement.

2. COMBUSTION CHAMBER RING.

The combustion chamber ring contains all the mechanism for carrying out the actual combustion process. It includes all the combustion chambers arranged in a circle, the compressed gas chamber, the piping, etc., for the air and cooling water (if water cooling be employed), also the gas and air inlet valves and the nozzle valves. The nozzles are fixed in front of the discharge passage for the combustion chambers. The reverse blades are also bolted to the top portion of the combustion chamber ring. The upper flange carries the pedestal, and the turbine cover is bolted to it and connected to the exhaust pipe.

The combustion chambers are preferably of elliptoidal form. They are flattened at the bottom and narrowed down at the top for the nozzle valves. In the bottom portions are the openings for the air and gas valves.

For this reason it is advisable that the part of the annular chamber from which the gas and air are delivered should be cast with the combustion chambers formed on the bottom side of a closed ring chamber.

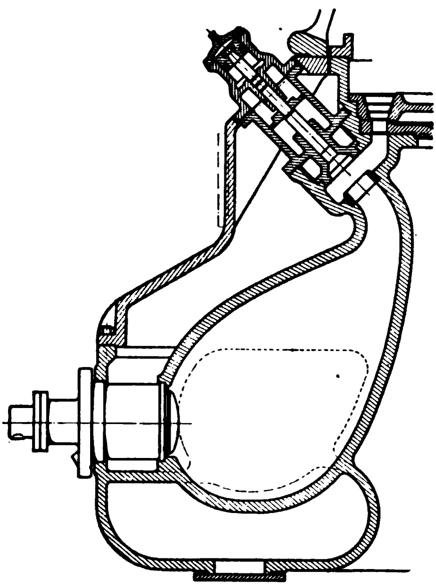


FIG. 74.—Combustion chamber ring.

The turbine casing, except the cover, the passages from the chamber to the nozzles, including the nozzle valves, the chamber ring flange, and the external jacket, can be cast in one piece with the base; or the base-plate may be dispensed with.

The top flange may also be in two parts, and the external shell bolted on a flange above the air valves; the turbine casing and cover are then free to expand independently of the shell For this purpose the nozzle valves must be bolted direct on to the combustion chambers, and not on to the shell, and clearance space must be allowed between them and the shell (fig. 74). Or the external shell and turbine casing can be cast together and bolted on; between the nozzle valves and the combustion chamber, expansion pieces, glands, etc., are inserted, which allow a certain play in the upper portions of the combustion chambers relative to each other and the turbine casing (fig. 75).

Another alternative is for the shell to extend only to below the nozzle valves (flap, piston, slide, or lever valve); the pedestal support between the combustion chambers and the dynamo rests upon this as before, and the turbine casing with the nozzle valves is inside the pedestal, but not connected with it, being either cast on or bolted to the combustion chambers (fig. 76).

It may be preferred not to draw the air from the turbine shaft or direct from the engine-room, but to provide a means by which the actual combustion chamber is cooled by circulating water, and in the section on the "Theory of the Gas Turbine" it is shown that the temperature of the mixture immediately before ignition should be as low as possible. For this purpose the combustion chamber down to below the nozzle valves may be arranged with a double wall, and the intermediate space cooled by water circulation; or the external shell can be utilised as the outer jacket, and an inner shell provided.

In the construction cast iron is to be preferred. The pressure is not nearly so high as in a reciprocating engine (10 atmospheres instead of

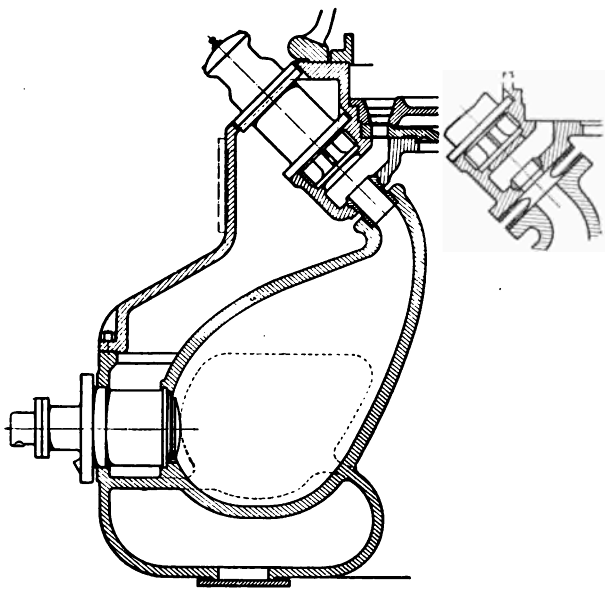


Fig. 75.—Combustion chamber ring with expansion pieces.

30 atmospheres), and pre-ignition can give rise to no danger, owing to the automatic opening of the nozzle valves in such a case. Moreover, the spherical or elliptoidal form of the combustion chambers is much more suitable for resisting the explosion pressure than a plain cylindrical construction, and it is clear for these reasons that the strength of the combustion chamber ring involves fewer difficulties than is the case with a cylinder, and that relatively thinner walls are required.

As a proof, it may be mentioned that the combustion chamber ring for a 1000-H.P. gas turbine weighs about 17 tons; the whole turbine weighs about 25 tons, which is very different from the weight of a 1000-H.P. reciprocating gas engine, this being about 140 tons without the flywheel.

The construction of the combustion chamber ring and turbine casing naturally depends on the point as to whether the whole is cast in one piece or bolted together as discussed above.

In the first case the machining is limited to the valve openings, the top flange, and the turbine casing, all carried out in the boring machine or lathe; and this is particularly simple with the use of the American method of setting up large castings on big turntables, on which the portable machine tools are arranged, and this method is certainly becoming more widely adopted in German factories. As the same work

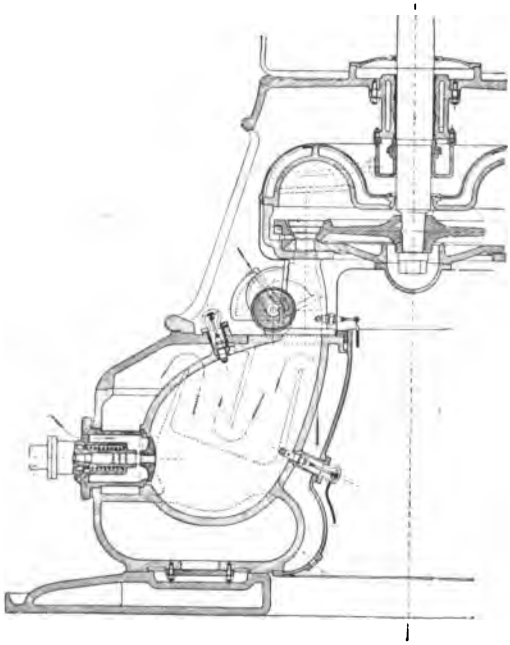


FIG. 76.—Gas turbine of 1000 H.P. with nozzle valves of vertical design.

has to be done in a number of combustion chambers, special boring and turning machines for accelerating and cheapening the work soon repay their own cost.

In the second case, such other boring and turning is required as is necessitated by the parts being divided instead of cast in one piece.

With horizontal turbines, the supporting feet of the combustion chamber ring, by which it is attached to the bedplate, have also to be machined.

The repetition of the individual parts of the combustion chamber ring simplifies the patterns necessary for their casting, as will readily be understood.

3. GAS AND AIR VALVES.

The air and gas valves serve the purpose of regulating the delivery of the air and gas respectively to the combustion chambers. There are therefore as many valves for each as there are chambers. Owing to the number of these valves, which in themselves are very simple, special care should be taken in the simplicity and reliability of the mechanism

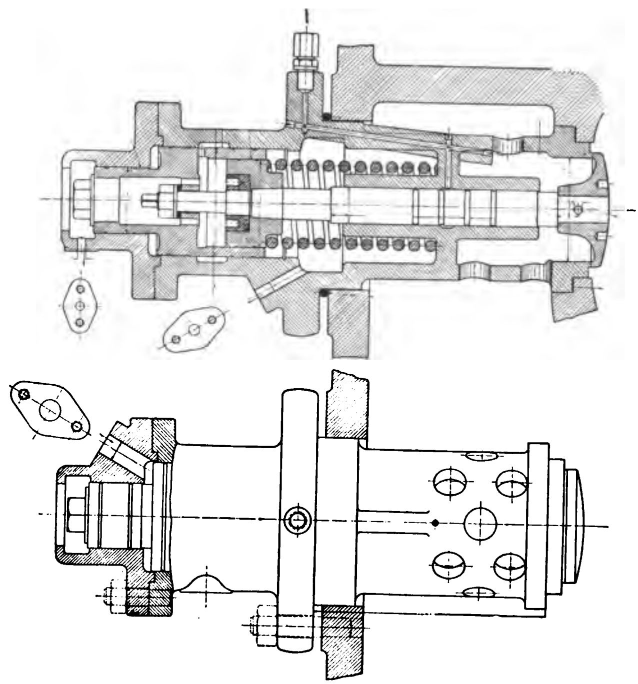


FIG. 77.—Section through gas valve.

actuating them. It is indisputable that the simplest and most reliable method lies in the adoption of a hydraulic arrangement with oil, and not in the employment of the mechanical gear commonly used with reciprocating engines. By this former means, levers, bolts, eccentrics, etc., are completely dispensed with, and all that remain are the piston valves, operated by oil under pressure provided by a rotary oil-distributing pump.

The construction of the oil-operated valves is not particularly complicated if arranged with a "hit-and-miss" type of governor also actuated by oil under pressure, whose supply is likewise regulated by the governor.

The valves for oil operation and regulation are generally arranged in valve casings bolted on the narrow end of the combustion chamber ring. The valve face bears on the seat of the casing; the valve spindle works in a bush in the casing, and carries a piston at the other end. The piston moves freely in another hollow piston, which itself works in the valve casing, actuated by oil under pressure. It communicates its motion to the valve piston when the space between the hollow piston and the valve piston is filled with pressure oil.

The stroke of the valve is limited by stops on the casing and

FIG. 78.—Details of gas valves, from a photograph.

the spindle. The valve piston bears against the spring fixed in the casing.

If it is desired to control the valve motion from the outside, the hollow piston should be designed as a differential piston on the side on which the oil is under pressure. If this regulation can be dispensed with when the turbine is running, it is sufficient to allow the oil pressure to be exerted on the whole surface of the hollow piston. The piping in the cover is then unnecessary (fig. 77).

The various parts of a gas valve are shown in fig. 78. A greater simplicity could scarcely be desired.

It may be mentioned that if, during the gas admission, pre-ignition occurs, there is no exceptional stress caused in the gas valve. As soon as the pressure of the gas on the valve face becomes greater than the oil pressure on the hollow piston, the valve closes automatically.

4. NOZZLE VALVES.

The main purpose of the nozzle valves is to close the combustion chamber during the admission of gas. They must be suitably arranged to be opened automatically by the force of the explosion; during the expansion which follows, and the scavenging interval, they must be kept open by oil pressure.

The opening of the nozzle valves may be controlled positively, but with this arrangement the property of the automatic opening by the explosion is not taken advantage of, and in the event of pre-ignition, the combustion chamber has no safety valve. Moreover, it is a

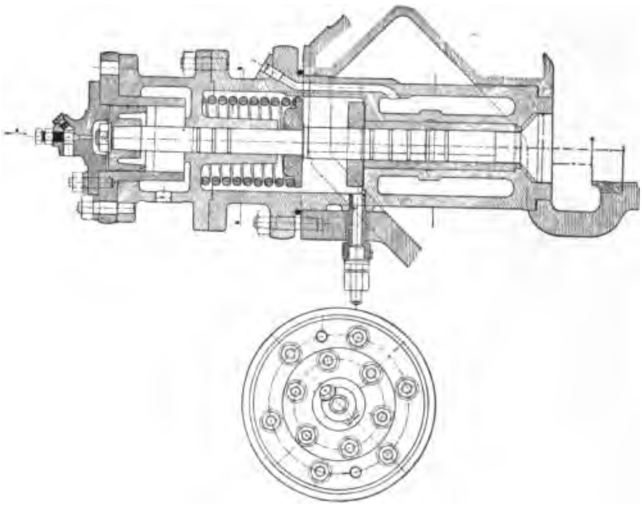


FIG. 79.—Section through nozzle valve.

question of some difficulty, with the positive valve, to arrange for the opening to be so rapid, exact, and flexible that the period between ignition and expansion is always a minimum whatever be the gas mixture, and the temperature of the mixture before ignition. This time period should undoubtedly be minimised in view of the necessary time occupied by the explosion itself, in order that the radiation losses may be maintained within such limits as to allow the combustion process to occur as a mechanical cycle with good efficiency. And it must be remembered that if the nozzle valve opens too early the actual combustion process is destroyed.

The nozzle valve, apart from the actual casing, consists of the valve and spindle, the spring collar and spring, and the control piston, which works in the cylinder containing oil under pressure. The spring keeps the valve on its seat in the combustion chamber when closed, under such a pressure that there is no escape during the admission of gas. After ignition, pressure oil enters below the piston, which it forces outwards, and consequently opens the valve, which it keeps open until scavenging is complete.

The oil under pressure is admitted below the piston of the nozzle valve a short time after ignition, whether the valve is opened by the explosion or not. The delivery of the oil is therefore not regulated by the movement of the piston. This has the disadvantage that, if the gas fails to ignite, the unburnt gas mixture remains in the chamber and is not scavenged out. If the turbine works in conjunction with an exhauster, no fresh air enters for the following cycle, but the air and gas mixture flows into the air space, and more gas is added to the mixture, which thus becomes richer than it should be for normal operation. A further failure to ignition would probably occur, and the result would be that the combustion chamber would no longer come into operation.

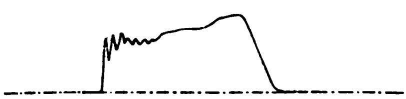


FIG. 80.—Nozzle valve lift diagram.

Fig. 79 shows a section of a single nozzle valve, and fig. 80 is a nozzle valve lift diagram on a time basis.

This shows clearly the opening of the valve (the vertical rise of the line representing the lifting of the valve) and the way in which it is held open by oil pressure, also the less rapid drop with the mechanical closing of the valve.

The nozzle valve may be replaced by other suitable arrangements, *e.g.* piston valve, Corliss valve, or flap valve. A piston or Corliss valve may also be opened by the pressure of the exploding gases. It is only necessary to employ a subsidiary piston, which actuates the piston or Corliss valve under the influence of the explosion pressure.

A flap valve for closing the combustion chamber can be made in a very simple form if, as is shown in fig. 81, its rotary motion is controlled on one side by a vane piston and on the other by a flat spiral spring. This arrangement is particularly useful inasmuch as it occupies a comparatively small space and gives a large passage area. It is evident from the section on the "Theory of the Gas Turbine," given previously, that this, in conjunction with a large nozzle opening, is much to be desired. There is the further advantage that the gases are not diverted in their passage to the nozzle, as is the case with ordinary valves.

On the other hand, the disadvantage of the flap valve, that it does not form such a perfectly tight joint as the other types, is not particularly important. This drawback is largely minimised by the fact that for a complete mixing of the gases, which is of the utmost practical import-

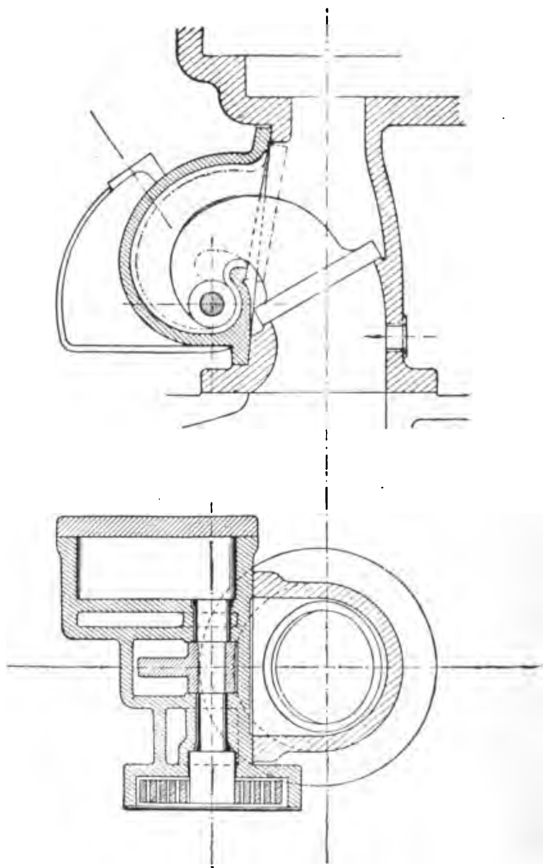


FIG. 81.—Section through nozzle flap valve.

ance, the nozzle valve should be slightly raised off its seat during the admission of the gas, or a portion of the admission.

A rigid connection between the valve spindle and the piston or the flap valve spindle and piston is not absolutely essential. Under certain conditions the valve or flap then allows a passage to the gases momentarily if the spindle has some play relative to the piston, and the connection is only rigid when the valve is open or mechanically closed.

5. VALVE OPERATION.

It has already been stated that the valves may be conveniently operated by means of oil under pressure. The actual control is in this case derived from the oil-distributor, and this can most simply be constructed of a rotary type.

For each set of valves, combining all the nozzle valves, air and gas valves, there is a rotor. These rotors are fixed separately or together on a spindle which has an angular velocity corresponding to the number of cycles per chamber.

Axial ports are arranged regularly in the stator of the distributor, communicating with the oil pressure spaces in the valves. The oil is pumped inside the rotor under pressure; it then flows through the ports in the rotor to those in the stator, and thence to the valves, so that these are actuated in a regular and predetermined order. When a particular port in the rotor is uncovered again, the oil flows back (owing to the pressure of the valve spring into the upper portion of the distributor), and thence to the pump.

This method of regulation can of course be modified. Oil can be delivered continuously to the valves, and when they are opened, the flow of the oil through the distributor can be stopped. On the valve closing, the oil once more flows freely through the distributor.

For this purpose the regulating oil supply pipe may be utilised, so that there would be a connection between the governing oil space and the chamber for the oil actuating the valves, and the latter would also be in communication with the distributor.

The exactness and rapidity of the lift of the valves with oil control is shown in the valve lift diagrams, figs. 83 and 84, for an air and a gas valve respectively, taken by means of a time indicator.

It was, of course, necessary in designing an oil-controlling arrangement to exercise considerable care to avoid pressure waves, shocks, and air pockets.

These early difficulties, which can easily be overcome by experience, are of small account compared with the endless advantages over mechanical valve gear, from the point of view of design, external appearance, and above all absolute reliability of operation.

The method of actuating the valve gear by oil necessitates a very reliable oil pump. As the shaft is most conveniently arranged to be supported by oil under pressure in a footstep bearing, it is of the utmost importance that this pressure be maintained. If the footstep and the distributor be provided with oil from the same pump, the supply of energy to the gas turbine is automatically cut off immediately the oil

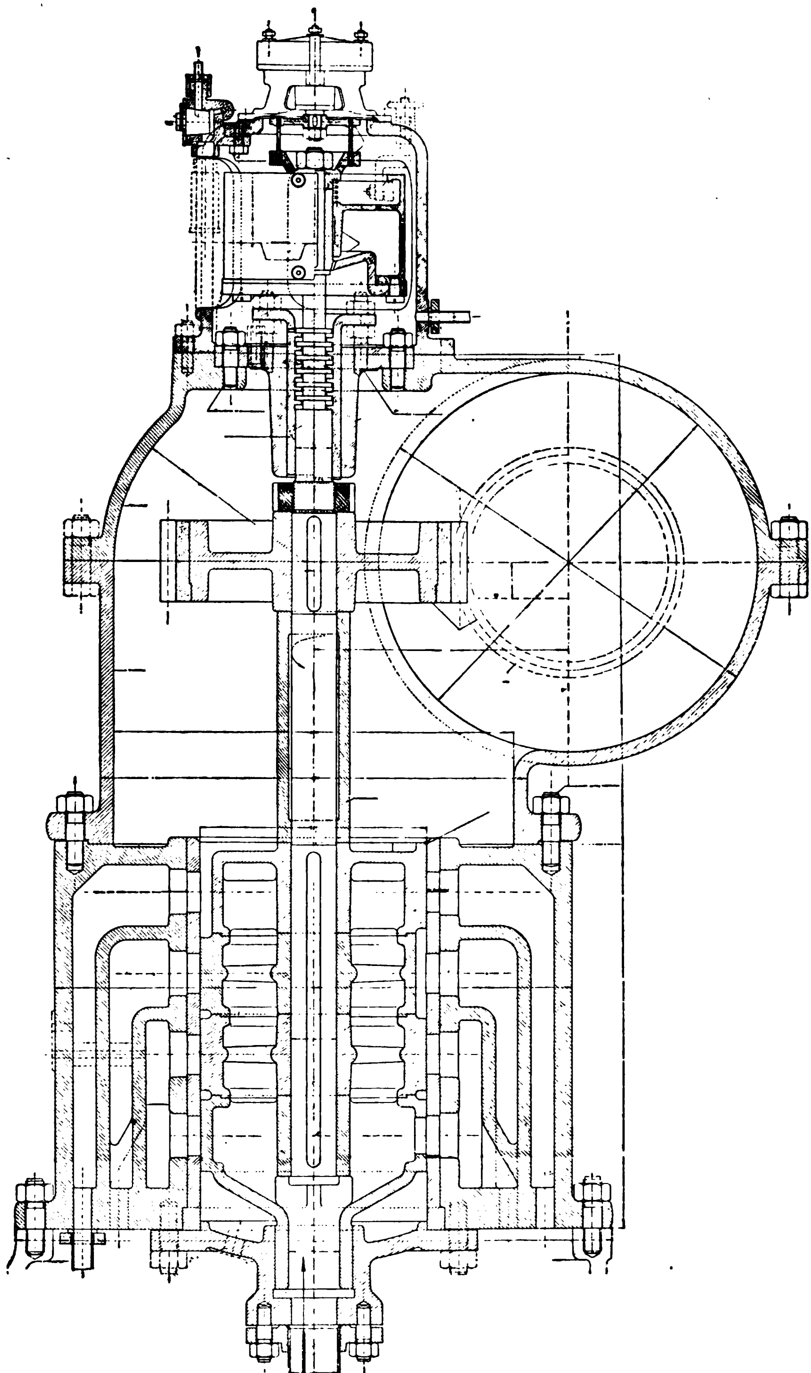


FIG. 82.—Section through rotary oil-distributor for actuating the valves.

pressure fails; but if a reserve supply of oil for the footstep be maintained by an accumulator and auxiliary piping (at least until the turbine

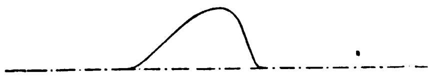


FIG. 83.—Lift diagram for air valve.

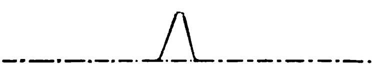


FIG. 84.—Lift diagram for gas valve.

stops, or the speed has dropped very considerably) there is no danger occasioned to the turbine if the pump be accidentally put out of action.

Fig. 82 shows a rotary oil-distributor for a 1000-H.P. gas turbine.

6. GOVERNING.

The principle of the method of governing a gas turbine is explained in the section on the "Theory of the Gas Turbine."

In the design of our first turbine for driving purposes, the "hit-and-miss" method combined with throttle regulation was employed. The governor shaft is driven from the turbine shaft, as explained above, by means of helical gearing, and on this shaft the governor is fixed, the sleeve being suitably connected to the horizontal floating lever of the actual governing arrangement. This, in principle, consists of an auxiliary motor operated by oil under pressure, the throttle, and the "hit-and-miss" regulator.

The auxiliary motor is of the usual form, and consists of the connecting rod, piston valve, and the actual motor. The piston rod bears at the bottom on the spindle of the throttle valve, but is not rigidly coupled to it. At the upper end the piston rod carries a small regulating piston for the "hit-and-miss" governing.

The throttle valve differs from the ordinary design in that it is opened by a spring; the throttle valve spindle moves with the auxiliary piston rod for only a portion of its stroke, since the stroke required for the "hit-and-miss" regulation is rather long, and it would be unnecessary to allow the same stroke for the throttle valve.

The "hit-and-miss" regulator is arranged so that the motion of the small controlling piston is varied by the servo-motor piston rod. It moves in a bush in the side of which are holes communicating with the space for the regulating oil in the gas and air valves.

Below the small piston is oil under pressure, and above it the oil can escape. The higher the piston rises, the larger the number of ports which give access to the oil under pressure, and the more chambers come into operation.

The whole governing gear is fixed on the top of the throttle valve, and this is bolted to the combustion chamber ring of the turbine.

Above the main governor there is a safety governor on the governor shaft. If the maximum speed be exceeded, ignition momentarily ceases. The safety governor can be arranged to close the throttle valve or cut off the oil pressure. If the gas turbine be driven by power gas, and there is not a substantial reservoir with a floating valve between the turbine and the blower, the sudden closing of the gas inlet valve may lead to trouble in the gas generator. The momentary stoppage of the oil pressure is also more disadvantageous than if ignition is shut off.

The governing gear is illustrated in fig. 71.*

7. IGNITION AND TIMING.

For the ignition of the gas mixture in the chambers a high-tension ignition apparatus is employed. As the principle of this type is described in detail in the pamphlets of Messrs Robert Bosch of Stuttgart, it is unnecessary to explain it very fully in this volume.

An accumulator ignition must be provided to bring the turbine up to full speed, as the magneto does not come into action until the turbine is run up. It may, however, be noted that the distance of the electrodes in the sparking plug should be as large as possible, in order to assure reliability of operation; the larger it is, the less is the danger that the plug may be put out of action through becoming dirty. But this distance is dependent, first, on the voltage of the magneto machine, and secondly, on the pressure of the gas and air mixture which has to be ignited by the spark. With the same voltage, the distance between the electrodes may be made greater the lower the compression.

In this particular the gas turbine with low compression has an advantage over the reciprocating gas engine with high compression, which is minimised by the fact that the employment of a mechanical "hit-and-miss" method of ignition is very unsuitable for the gas turbine.

As no oil is used in the combustion chamber, there is no danger of miss-fire through an excess of oil.

For the timing of the ignition and the oil valve control, a small drum rotating with the oil-distributor spindle is employed. If an indicator is arranged in the oil piping to the air, gas, and nozzle valves, and the oil pressure is drawn on a card on the drum, a diagram of the variation of pressure of the oil is obtained at the point where the indicator is fixed. From the relative positions of the diagrams to each other, the intervals between the various phases of the cycles can be controlled.

* A variation from this method of governing may be made by taking the oil to the governing gear direct, thence to the valves, which are then quite simple. If a chamber is to be cut out, the supply of regulating oil to the governor gear has simply to be shut off.

If needle electrodes be fixed to the indicator pencil connected to the corresponding contacts of the igniter, the sparks make small holes in the paper, which correspond with the ignition in the chambers relative to the oil pressure diagram.

It is only possible to obtain very accurate results with the time indicator when it is attached directly to the oil pressure space in the valve, or when a valve lift diagram is drawn with it. The time of ignition is obtained in a similar way.

This arrangement is shown in fig. 82.

8. AUXILIARY APPARATUS.

The regenerator for the production of steam for driving the blowing plant was mentioned in the section on the "Theory of the Gas Turbine" as a requisite auxiliary for a gas turbine. It consists of a shell for the exhaust gases, a coil of water piping, and a steam collector.

As is shown in fig. 85, the exhaust gas enters the regenerator through an opening in the top; it is distributed in the annular space, and flows with the necessary velocity over the long coil of pipes (one section being small) to the lower annular space, where the temperature of the gas has fallen to about 100° C., and it then passes into the exhauster, or into the open air if the compressor arrangement be adopted.

The feed-water enters the coil from below, and begins to evaporate in the middle of its passage through the pipes. The steam is collected in the steam collector, placed directly above the water piping. It is taken to the blowing plant direct from the steam dome. The coil for the water can be arranged with a gland fixed directly in the shell surrounding it.

The whole regenerator can be supported from the top annular portion of the shell. The S-shaped exhaust pipe is connected to the turbine through a flexible expansion piece.

To provide for all contingencies, spring-loaded safety covers are arranged on the top and bottom annular chambers of the shell. If through any unforeseen circumstances, such as negligence on the part of the attendant, the gas and air mixture should enter the regenerator and be ignited there, the regenerator would not be damaged, since the consequent rise in pressure would soon be dispelled by the large safety openings.

It must be admitted that the design of the first regenerator was fraught with considerable difficulty, as similar plants are not often used. The comparatively slow delivery of the heating gases to steam boilers cannot be allowed in a regenerator; the temperature of the gases in the former is much higher, and the transmission of heat to the water therefore takes place chiefly by radiation; transmission by conduction only comes into consideration in relation to the radiated heat when the gas has already been cooled down; for this reason, in steam boilers a steady flow of the heating gases is not of fundamental importance, as in a regenerator, where the gases are relatively cool.

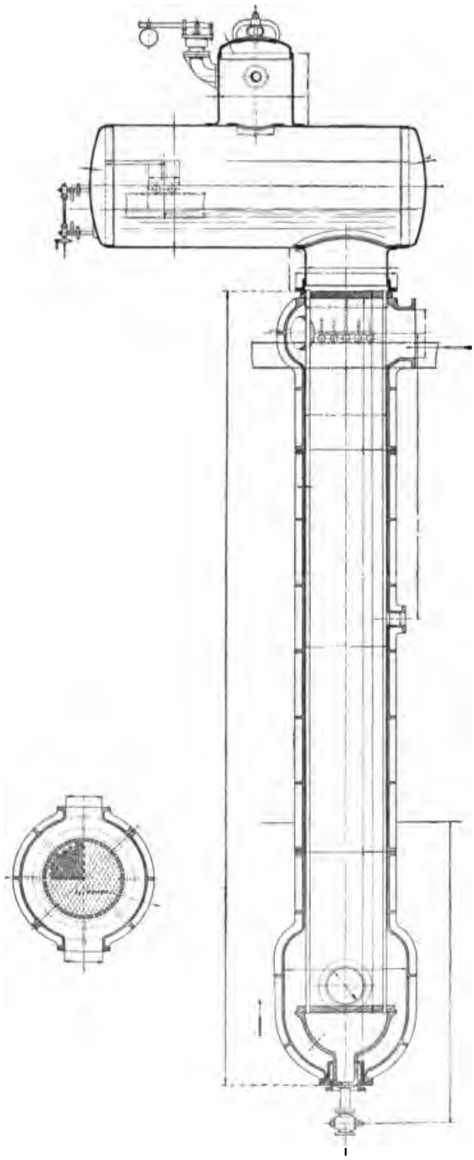


FIG. 85.—Section through regenerator.

It is quite useless to base the design of a regenerator on that of a steam boiler, and to do so would lead to impossible dimensions for the regenerator.

The blowing plant consists of a blower, the air circulator (in our case an exhauster), and a steam engine, mounted on a bedplate.

The blower and exhauster are of the turbine type, and their construction has frequently been described. The gas blower contains a series of pressure stages arranged sequentially, and the axial thrust is taken up by equalising pistons. The neck glands are connected on the pressure side with the suction space in the blower. In this way it is possible to prevent completely any leakage of gas into the engine-room, and it is only necessary to ensure that the pressure in the suction space does not exceed a certain figure.

The very simple construction of the blower allows ready inspection of the wheels for the purpose of clearing them of dirt, dust, and tar.

The exhauster is arranged as a double-flow machine, and axial pressure is thus prevented, so that no equalising pistons or similar arrangements are necessary. The construction is extremely simple.

Enough has already been said regarding the steam turbine for driving the blowers. It is provided with a governor to give constant speed, and a regulating arrangement to permit of constant gas pressure or steam pressure in the regenerator steam-collector.

PART III.

GENERAL COMPARISONS.

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1. COMPARISON OF THE GAS TURBINE AND THE RECIPROCATING GAS ENGINE TREATED THERMO-DYNAMICALLY.

UP to the present the reciprocating gas engine has only casually been mentioned in comparison with the turbine, since the gas turbine cycle can be developed *per se* without analogies, and moreover the construction shows no relation to that of the reciprocating engine.

A comparison between the gas turbine and reciprocating engine is, however, of considerable interest, not only from the point of view of the turbine, but also in connection with the development of the reciprocating gas engine. The history of this development has often been discussed. The various important stages are very clearly shown in fig. 86, following Schlöttler's comparison in *Die Gasmaschine*, 1909. In this figure the years are drawn as abscissæ, and the ordinates are the efficiencies of the reciprocating gas engines, based on gas per effective H.P., at the various epoch-making stages of development.

Naturally, the absolute figures cannot be compared exactly with each other, as the tests were made by various authorities, at different times, and under different conditions. Moreover, this representation takes no account of the development of the reciprocating engine from the aspect of actual operation, and has no reference to the purpose of each particular machine.

If these limitations be borne in mind, it is evident that the illustration is accurate if one wishes only to see how the economy of the reciprocating gas engine has progressed. The figure begins in the year 1860 with Lenoir's engine, having an efficiency of about 4 per cent. In 1867 the free-piston engine was produced by Otto, with about 15 per cent. efficiency. In 1878 there was Otto's four-cycle engine, with about

12 per cent. efficiency; in 1897, the Diesel motor of about 25 per cent. efficiency.

The four-cycle motor alone has been developed to any considerable extent. Parallel with this development, the two-cycle engine has progressed with the same limits of combustion and compression. In 1888, at the Exhibition in London, efficiencies of 15–19 per cent. were obtained with four-cycle engines running on lighting gas. In the years 1895–1905 the efficiency of the four-cycle motor working on power gas was raised to 16·5–20 per cent., and to-day the efficiency under these conditions is reckoned at 20–24 per cent.

The efficiency of the Diesel motor has been increased from 25·2 per cent. in 1897 to about 31–33 per cent. at the present time.

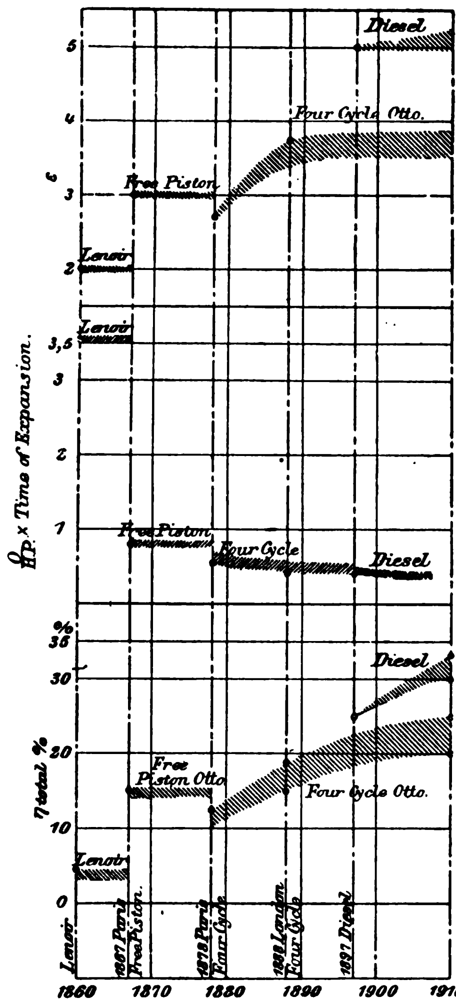

FIG. 86.—Curves showing historical development, η, ϵ, $\frac{O}{H.P.}$ × time.

It is often stated that the gas turbine cycle approximates to that of the Lenoir engine; this does not show the gas turbine in a very favourable light, as the efficiency of this engine has remained very low. The cycle of the Lenoir engine will therefore be shortly recapitulated.

The engine is a double-acting machine; the gas mixture is drawn in, and ignited at the centre of the stroke. The pressure rises to 5 or 6 atmospheres, and the gases expand down to 1·5–2 atmospheres, and are then expelled.

According to Schlöttler (page 389), the Lenoir cycle is in itself no less efficient, under the same conditions, than that of the ideal free-piston reciprocating engine, if no account be taken of the variation of specific heat, nor of the heat given up to the walls. In fact, it is somewhat better, being 29 per cent. against 25 per cent.

In his discussion on large gas engines,* Riedler states that the actual efficiency of the Lenoir engine, apart from mechanical deficiencies, is so low because the expansion ratio is small (no compression, small expansion), and also because the amount of heat transmitted to the walls is relatively large, due to the low piston speed and the comparatively long time the gases are in contact with the cylinder walls.

If the Lenoir cycle be calculated in a similar manner to the gas turbine cycle, particularly with reference to the heat radiation to the large surface of walls (much larger than in a four-cycle engine) at the moment of maximum temperature, it would be seen that an ideal Lenoir engine would not have an efficiency many times greater than that actually obtained. The imperfections of the practical engine are thus not so great, since in the calculation the important factors (heat radiation) are included.

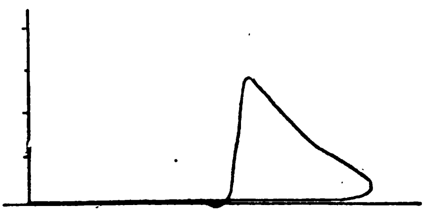

FIG. 87.—Lenoir diagram.

It must be considered, moreover, that the ignition in the centre of the stroke is at least as inefficient as late ignition in a four-cycle motor, since the first ignition wave follows the piston (the speed of which is at a maximum at the centre of the stroke) and ignites relatively slowly, with a tendency to fail altogether. The factors leading to low efficiency may therefore be determined not only generally but individually, and this is quite necessary if it is desired to arrive at a complete comparison.

It is perhaps of interest in the comparison to determine the product of the area of the combustion chamber per H.P. multiplied by the combined times of explosion and expansion. This figure gives approximately how much heat is radiated to the walls relative to the heat supplied. For an exact comparison the cycles must be calculated right through.

According to Schlöttler, the data for the Lenoir engine tested were as follows:—

Diameter, 160 mm.; stroke, 300 mm.; $n=81$; piston volume, 6 litres power about 1 H.P. (eff.).

The area of the combustion space at the end of expansion is 19·2 sq. m., and the ratio

$$\frac{O}{\text{H.P.}} = 19{\cdot}2 \text{ sq. m.}$$

* *Z. d. V. d. I.*, 1905, p. 316.

The time of duration of expansion was 0·185 sec., and therefore

$$\frac{\mathrm{O}}{\mathrm{H.P.}} \times t = 19{\cdot}2 \times 0{\cdot}185 = 3{\cdot}55.$$

The expansion ratio = 2.

The working diagram of the Otto free-piston engine (fig. 88) is as follows:—

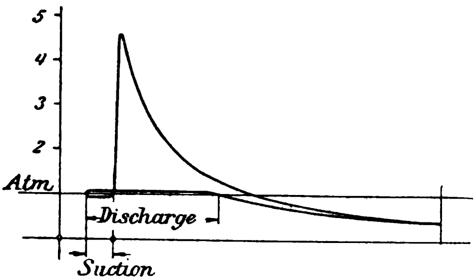

FIG. 88.—Otto diagram.

Ignition takes place much in the same way as with the Lenoir engine. As the piston moves upwards, ignition takes place after sufficient gas and air mixture has been drawn in. This corresponds to late ignition in modern four-cycle motors. It is well known that this is very unfavourable from the point of view of heat efficiency, owing to the incomplete combustion. The published tests of Riedler * show that, owing to late ignition, the indicated efficiency may drop 50 per cent. The reason lies in the extremely complicated actions during the combustion.

From this point of view, therefore, the free-piston engine possesses no advantages thermo-dynamically.

On the other hand, the expansion ratio is greater. Naturally, in the calculation it must not be taken that the final pressure attained when the piston reaches the end of its stroke is alone of importance in the cycle. If the question of the heat exchange with the walls be neglected, the free-piston reciprocating engine would work thermo-dynamically equally well if the stroke were only one-half, and the kinetic energy of the piston were stored up by an air-buffer or a strong spring for the outward stroke.

In this case the expansion ratio is approximately

$$= \left(\frac{p_1}{p_2}\right)^{\frac{1}{1{\cdot}37}} = \text{about } 3.$$

For the above condition to obtain exactly, the piston must impart to the exhaust gases, on the outward stroke, exactly that amount of energy which they give to the piston in falling to a pressure below atmospheric. This is necessary in order that the gases may again reach atmospheric pressure.

* *Z. d. V. d. I.*, 1905, p. 316.

On the outward stroke, therefore, under the above conditions, only the energy from expansion to atmospheric pressure can be given up.

If the indicated work is actually increased by expansion below the atmospheric line, this is solely due to the influence of the walls. During the comparatively slow outward stroke of the piston, the exhaust gases are greatly reduced in temperature. The compression of the exhaust gases from the final pressure to atmospheric pressure is therefore accompanied by a large absorption of heat; the compression line is very flat, more so than the previous expansion line. For this reason the second portion of the stroke acts not only as an air-cushion, but on the outward stroke develops useful work.

This is a well-known instance of the fact that a consideration of the cycles of combustion and explosion engines without reference to the heat passage to and from the walls is to a certain extent valueless.

The gain in indicated work by expansion below the atmospheric line varies, and the speed of revolution of the engine is naturally of considerable importance in this particular.

If it be taken that this gain is an actual addition, and the Lenoir engine be then compared with the better free-piston engine with expansion down to atmospheric pressure, the combustion and expansion down to this atmospheric line are practically only more advantageous in the latter case in so far as the time of combustion and expansion is much less than in the Lenoir engine. As there is no pre-compression either in the Lenoir or the free-piston engine, the ratio $\frac{\text{O}}{\text{H.P.}}$ only differs as the efficiency varies. The total duration of the stroke should amount to about 0·04 sec., and less than that when the atmospheric line is reached. Taking it at one-half, or 0·02 sec., the amount of heat given to the walls under similar conditions would be 0·185 for the Lenoir engine and 0·04 in the free-piston engine, and the ratio is—free piston: Lenoir = 1 : 10·8.

These ratios are therefore more favourable than with the Lenoir engine, without considering the additional work obtained by expansion below the atmospheric line, although the ignition is the same.

The product $\frac{\text{O}}{\text{H.P.}} \times$ time, with actual engines of the free-piston type, was obtained by Schröter (p. 289). The piston volume was 17·3 litres, piston diameter 150 mm., stroke 980 mm.; the surface of combustion space at the end of the expansion was

$$9{\cdot}8 \times \pi \times 1{\cdot}5 + 2 \times 1{\cdot}77 = 49{\cdot}7 \text{ sq. m.}$$

The power $= 0{\cdot}64$ H.P., and therefore

$$\frac{\mathrm{O}}{\mathrm{H.P.}} = 77{\cdot}6.$$

The time was $0{\cdot}04$ sec.; hence

$$\frac{\mathrm{O}}{\mathrm{H.P.}} \times \mathrm{time} = 77{\cdot}6 \times 0{\cdot}04 = 3{\cdot}11.$$

If the duration of expansion be considered only, the atmospheric line

$$\frac{\mathrm{O}}{\mathrm{H.P.}} \times \mathrm{time} = 0{\cdot}8.$$

Comparing this with an early Otto four-cycle engine of similar dimensions—*e.g.* Schlöttler, p. 292 (1886 engine)—in which the piston diameter was 170 mm., the stroke 340 mm, and the surface of the combustion space at the end of the stroke was $\mathrm{O} = 33{\cdot}44$ sq. m.,

$$\frac{\mathrm{O}}{\mathrm{H.P.}} = 3{\cdot}63.$$

The total duration of combustion and expansion was $0{\cdot}163$ sec., and hence

$$\frac{\mathrm{O}}{\mathrm{H.P.}} \times \mathrm{time} = 0{\cdot}592.$$

The approximate expansion ratio was $= 2{\cdot}7$.

A modern Deutz four-cycle motor was tested by Preuss in 1907 (Schlöttler, p. 295).

Piston diameter $= 184{\cdot}8$ mm.
Stroke $= 320$ mm.

The surface of combustion space at end of stroke was

$$\mathrm{O} = 27{\cdot}1 \text{ sq. m.}$$
$$\mathrm{B.H.P.} = 8{\cdot}57,$$

hence
$$\frac{\mathrm{O}}{\mathrm{H.P.}} = 3{\cdot}16.$$

The total duration of combustion and expansion was $0{\cdot}134$ sec., and

hence
$$\frac{\mathrm{O}}{\mathrm{H.P.}} \times \mathrm{time} = 0{\cdot}423.$$

The approximate expansion ratio was $= 3{\cdot}7$ with $p_2 = 4$ atmos., with a compression pressure of $11{\cdot}3$ atmos. and an expansion pressure of about 24 atmos.

The following data relate to the first small Diesel motor tested by Schlöttler (p. 321):—

Piston diameter = 250 mm.
Stroke = 400 mm.
Surface = 41·3 sq. m.
B.H.P. = 17·8
$\frac{O}{H.P.} = 2{\cdot}32$
Time = 0·195 sec.
$\frac{O}{H.P.} \times \text{time} = 0{\cdot}452$
$\epsilon = 5{\cdot}0$ approximately.

It is to be noted that all the foregoing tests were made with lighting gas, except the last, which was with petroleum; it is further to be kept in mind that all the cycles except the last are explosion cycles, whereas the last is a combustion cycle with constant pressure and lower temperature (c_p instead of c_v).

The following table (Table VI.) gives comparisons from the different points of view:

1. Ratio of the surface of the explosion or combustion space at the end of the expansion to useful work, multiplied by the time of duration of combustion and expansion.
2. Expansion ratio.
3. When ignition takes place.

TABLE VI.

	Lenoir.	Otto's Free-piston Engine.	Deutz Four-cycle (Otto).	Deutz Four-cycle Tested by Preusz.	Diesel.
Piston stroke . m.	300	980 490	340	320	480
,, diam. . mm.	160	150	170	185	250
Revs. per min. . .	81	...	184	224	154
Power . B.H.P.	1	0·64	9·2	8·57	17·8
Surface O . sq. m.	19·2	49·7 24·85	33·44	27·1	41·3
Duration of combustion and expansion. . . . sec.	0·185	0·04 0·02	0·163	0·134	0·452
$\frac{O}{H.P.} \times$ time . . .	3·55	0·8	0·592	0·423	0·452
Expansion and ratio ϵ	2·0	3·0	2·7	3·7	5·0
Ignition	While piston goes forward.	do.	Before the dead point.	do.	Constant pressure combustion.
Actual η_{total} . . .	3·3	15·3	17·2	21·5	26·2
Year	1860	1867	1886	1907	1897
Fuel.	Lighting gas.	do. to atmospheric line.	do.	do.	Petroleum.

The tables, as was previously made clear, only shows that these conditions bear on the case of a combustion cycle. The figures must not be taken as absolute values (which can only be determined by careful calculation, as shown in the section on the "Theory of the Gas Turbine"), but as approximate relative values.

To be exact, these comparative figures would have to be derived from experiments on engines of equal dimension or power.

With this limitation, the table serves a useful purpose in that it gives in figures the increase of efficiency as the development of the reciprocating gas engine proceeded, even though these figures are approximate only.

If we compare the gas turbine (particularly for large power) with these stages in the development of the reciprocating gas engine, we have for the turbine

$$\frac{\text{O}}{\text{H.P.}} \times \text{time} = 0{\cdot}15 \text{ to } 0{\cdot}30,$$

and an expansion ratio of

$$\epsilon = 5 \text{ to } 6.$$

The ignition corresponds to ignition at the dead centre, and hence, bearing these facts in mind, it can certainly not be stated that the gas turbine cycle is similar to the Lenoir cycle; it is much more comparable with the cycle in the free-piston engine (which the Humphrey gas pump resembles). But even this comparison is not by any means exact. The gas turbine cycle is one fundamentally developed, just as the four-stroke two-stroke, or Diesel cycle.

2. COMPARISON OF THE GAS TURBINE WITH THE RECIPROCATING GAS ENGINE FROM THE POINT OF VIEW OF CONSTRUCTION AND OPERATION.

A modern 1000-H.P. double-acting, four-cycle, reciprocating gas engine dynamo set is shown in fig. 89, with a 1000-H.P. gas turbo-dynamo plant (including blower) drawn in thick lines. This illustration needs no elaboration or description.

The same gas turbine plant is also compared in fig. 90 with a two-stroke tandem gas engine dynamo of rather older design. Exact comparative data are given in Table VII.

In common with the steam turbine, the gas turbine has the advantage over the reciprocating engine that it is free from shocks and vibration; it requires no cylinder lubrication, and much less lubricating oil generally for its operation. The lubrication, as in the steam turbine, is more regular and certain than with the reciprocating steam engine, and the same holds with the gas turbine relative to the gas engine, as all moving parts run in an oil bath. The wear on these parts is therefore no more than similar parts in a steam turbine.

TABLE VII.—COMPARATIVE TABLE OF A GAS TURBO-DYNAMO OF 1000 H.P. AT 3000 REVS. PER MINUTE, AND A MODERN DOUBLE-ACTING TANDEM FOUR-STROKE 1000-H.P. GAS ENGINE DYNAMO AT 107 REVS. PER MINUTE.

	Gas Turbine.	Recip. Gas Engine.	Ratio.
Over-all dimensions: maximum length . . mm.	5,750	16,000	1 : 2·78
,, width . . ,,	6,100	7,200	1 : 1·18
,, height . . ,,	6,400	7,450	1 : 1·16
Area : maximum length × maximum width sq. m.	35	115	1 : 3·28
Weights: Actual machine . . . . . . kg.	25,500	140,000	...
Blowing plant . . . . . . ,,	10,500	...	...
Gas turbine and blowing plant . . ,,	36,000	...	...
Regenerator . . . . . . ,,	85,000	...	...
Gas turbine and blower and regenerator ,,	44,500	140,000	1 : 3·15
Dynamo rotor . . . . . ,,	1,600	40,000	1 : 25
,, stator . . . . . . ,,	5,100	42,000	1 : 8·25
,, complete . . . . . . ,,	9,000	84,000	1 : 9·35
Complete machine . . . . . ,,	53,500	224,000	1 : 4·2
Maximum dynamic force . . . . . . ,,	500 (peripheral pressure)	120,000 (piston rod pressure)	1 : 240
GD^2	200	1,200,000	1 : 6000
Total E . . . . . . m.-sec.$^{-2}$ kg.	8	30,000	1 : 3750
Flywheel moment $\frac{Ew^2}{2}$ . . . . m.-sec.$^{-4}$ kg.	395,000	1,910,000	1 : 488
Maximum degree of irregularity at full power . .	1 : 1000	1 : 400	1 : 2·5
Normal expansion pressure	8 to 10	20 to 25	1 : 2·5

The maximum amount of wear to be expected on the nozzles cannot yet be definitely given, in spite of the many experiments. As no water can form in the turbine, and the finely divided water particles are the greatest sources of wear on blades and nozzles, it is not to be anticipated that the wear in a gas turbine would be more than in a steam turbine.

There can be no trouble with acid so long as no water enters mechanically, and the water formed in the cycle is discharged only as highly superheated steam, and so long as the temperature of the exhaust gases does not fall below 100° C.

No compressed-air plant is required to start up the set, as in a reciprocating engine, but only steam to run the blower, and this is only needed until it is generated from the regenerator.

With reciprocating gas engines, 3000–4000 H.P. is the limit for a single unit. From the construction of the gas turbine, it can at once be said that the maximum output for a single unit is not dependent on the maximum allowable dynamic force, but on questions of transport; in a reciprocating engine design it is agreed that, when the piston-rod pressures exceed 300 tons, there are considerable difficulties involved. If in a gas turbine the combustion chamber ring be in parts, the

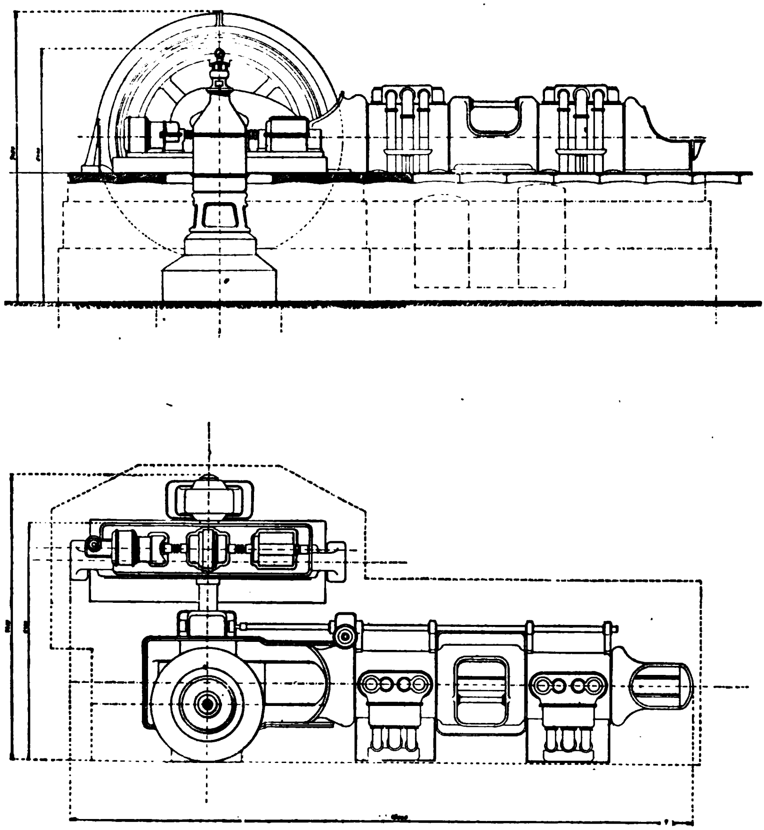

FIG. 89.—1000-H.P. gas turbine compared with a modern 1000-H.P. four-stroke reciprocating gas engine.

external diameter may be as much as 7 to 8 metres for railway transport; and with such sizes, powers of 8000 H.P. and more can be obtained in single units.

It is probable that the gas turbine will be developed particularly for large units.

With further experience, the number of cycles per chamber per minute will probably be increased, and this would allow a specific power per kilogram of material to be considerably raised.

3. GAS TURBINE PLANTS.

Only the experimental plant described on pages 69 to 89 and shown in fig. 92 and figs. 119–122 can now be cited as gas turbine plants, others being proposed arrangements.

Fig. 91 shows a gas turbine installation of three 1000-H.P. units and

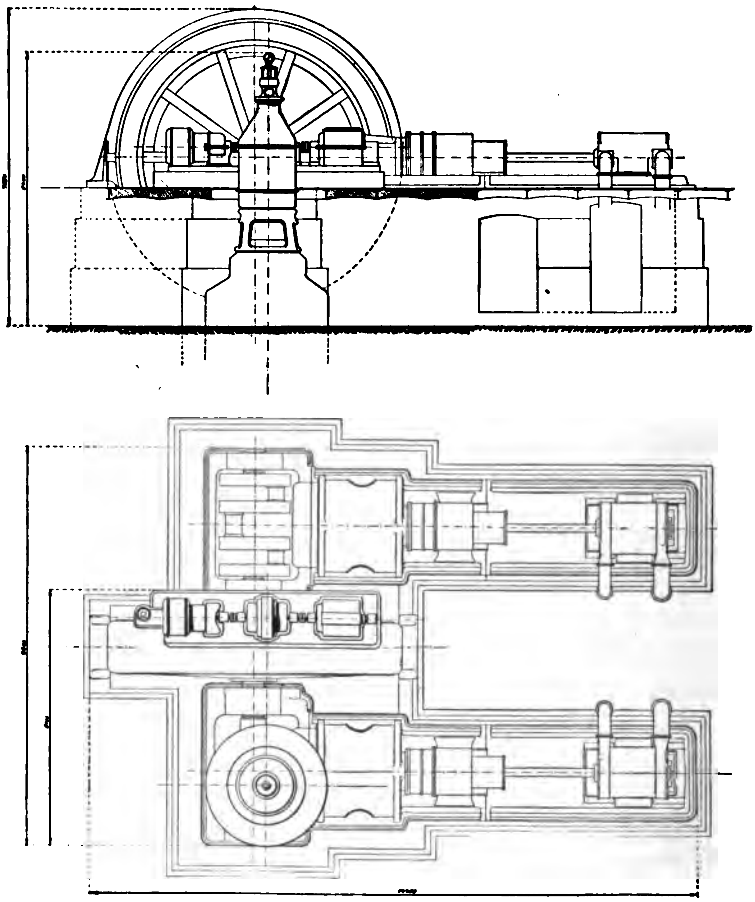

FIG. 90.—1000-H.P. gas turbine compared with a 1000-H.P. two-stroke reciprocating gas engine.

three 1000-H.P. power-gas generators, the turbines being of the vertical type. The blowers are on the same level as the dynamos, the gas turbines themselves and the piping being at ground level. At about the same height as the blowers and dynamos is the platform for the gas generators and the feeding arrangements for the coal. The regenerators are installed in the gas generator room, since they are controlled by the same staff as that for the generators.

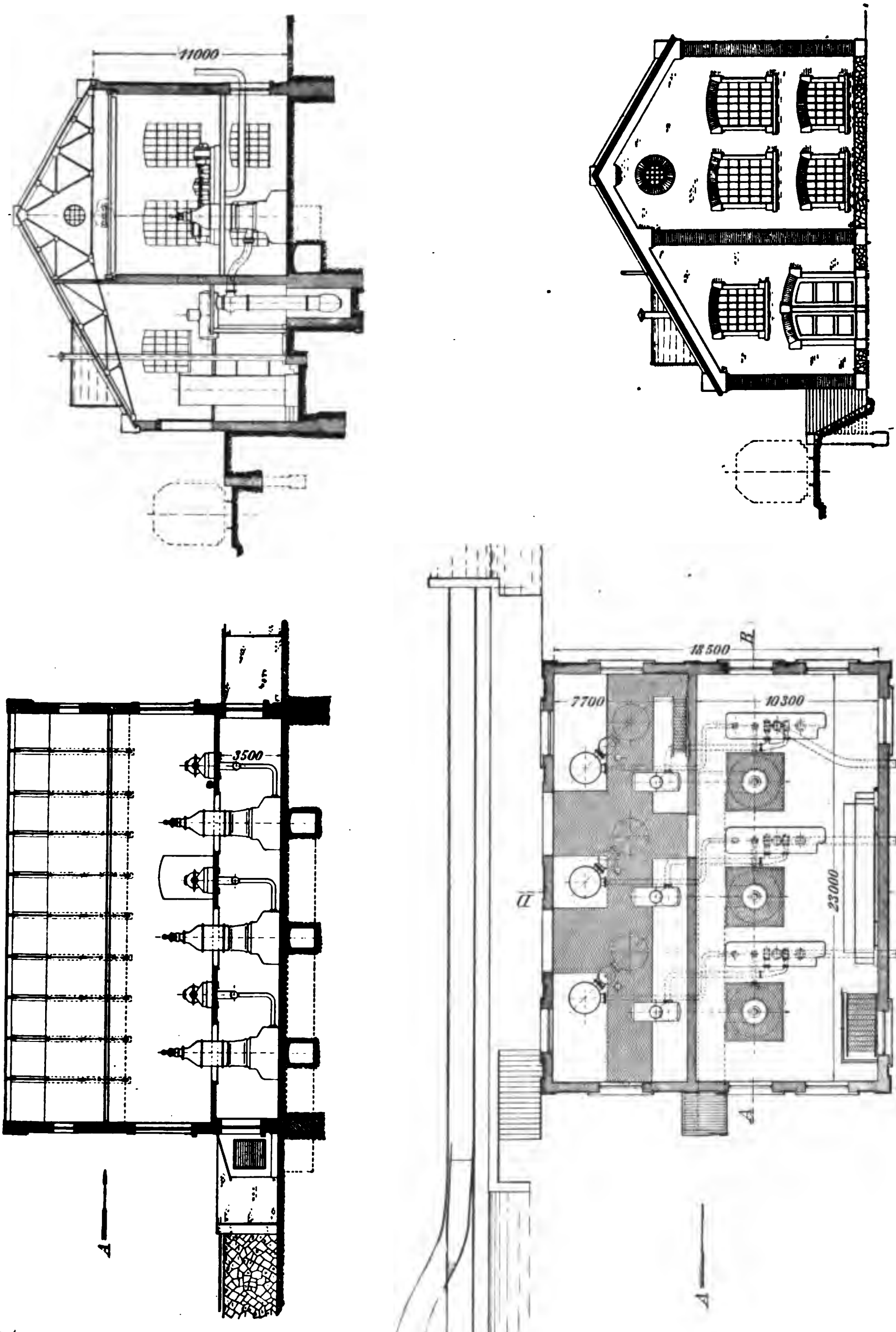

FIG. 91.—Three gas turbine units of 1000 H.P., including regenerators and power gas plant.

PART IV.

RESULTS OF TESTS.

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1. TESTS ON THE ORIGINAL GAS TURBINE.

FIG. 92 shows the first gas turbine, which was built in 1908 and tested in the winter 1908–9.

The scavenging, cooling, and combustion air was provided by a Root's blower, seen in the foreground on the left. After leaving the rotor, the exhaust gases and scavenge air are discharged into the atmosphere through the vertical pipe. When the engine was run on gas, this was delivered through the small delivery pipe on the right of the gas turbine, by a Root's blower. The blowers were driven by separate electric motors.

The experiments for running on oil were first made with one chamber. The Diesel type injection and pulverising valve can be clearly seen, in connection with the injection air vessel and the auxiliary compressor.

The top part of this turbine is in principle the same as that of the first gas turbine for driving a dynamo, and the photograph shows the arrangement.

It can be seen that the mechanical operation of the valves closely resemble that in reciprocating engine design. No control of the nozzle valve was provided originally. It was arranged only to retard the closing of the nozzle valve by means of a dashpot having a piston with small holes; but it was soon apparent that this would be unsatisfactory, and that a controlling mechanism must be provided.

The object of the tests with this first gas turbine was mainly to determine whether the cycle which it was desired to follow, could be satisfactorily carried out; whether the combustion could take place approximately as required, from the theoretical point of view; whether the scavenging and cooling were sufficient for continuous running of the turbine, without any mechanical troubles arising; and whether the blading was suitable for the cycle.

None of these questions could be definitely answered beforehand in the affirmative, and no single one amongst them could be settled by reference to particular experiments on, or analogies to, other engines.

Naturally, these points were much doubted on all sides, but there

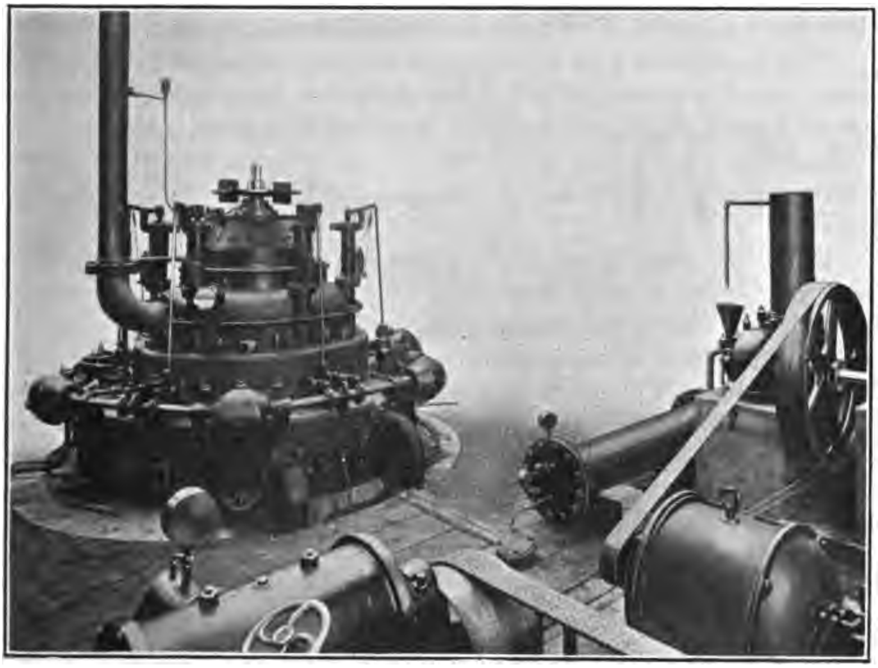


FIG. 92.—First gas turbine.

were some surprising results. The first difficulties were overcome with comparative ease, and after a few weeks the engine ran perfectly. The cycle was thus a practical one, there were no mechanical troubles, and the nozzles and blades showed no tendency at all to corrode, although the flame was allowed to reach the turbine rotor.

These tests were made with power gas from anthracite, and contained, therefore, 20 per cent. of hydrogen and some heavy hydrocarbons.

No modifications of an important nature were found to be necessary.

Opening the nozzle valves during a small portion of the gas-charging period was found in this motor to be a considerable advantage, if not absolutely necessary.

In the first experiments with power gas from anthracite, a make-and-break ignition was employed with the usual contact devices, actuated from the cam shaft driven off the engine. This arrangement necessitated that the sparking position should be made to suit the most convenient position of the contact rod, *i.e.* forward.

In the later experiments, this make-and-break ignition was replaced by high-tension ignition. With this method there may be any number of ignition points, in any desired position, and at any distance from the walls in the combustion chamber. With the adoption of this arrangement the combustion is very much more complete.

Only the following fuels were employed: lighting gas, motor petrol, heavy petrol, heavy motor petrol, petroleum, German gas oil, Roumanian gas oil, benzol, tar oil, and coal dust from various sources.

Lighting gas may be used with the same arrangements as power gas, except that the stroke of the gas valves must be considerably reduced.

Liquid fuel must be pumped up and injected by means of injection air and a Diesel type of pulveriser. The same ignition arrangements were needed with all the experiments (high-tension plugs).

No difficulty at all was experienced with lighting gas, as might have been expected. The large proportion of hydrogen and hydrocarbons contained in the gas allows a very rapid ignition of the whole contents of the combustion chamber.

It was found that liquid fuel could be employed and give complete and smokeless combustion without the use of ignition oil. It was only necessary to maintain the requisite mean temperature in the combustion chamber for each particular oil, to prevent an insulating film settling on the electrodes, and thus causing a miss-fire.

Benzine, heavy benzine, heavy motor petrol, and benzol could therefore be exploded in the cold turbine; the mean temperatures for various oils have to be approximately as under:—

400° C. for petroleum.
440° C. for gas oil.
475° C. for tar oil.

The possibility of obtaining complete combustion in the turbine with heavy oils, which it has only been possible to utilise in Diesel engines, was a pleasurable surprise, which may have very far-reaching results.

On the other hand, there was but little success with coal dust, although many kinds were tried. Only Cannel coal very rich in gas—and therefore very expensive—gave regular combustion; but only

20 per cent. of the coal was burned, the rest not having sufficient air for combustion.

It should be stated that the temperatures mentioned above as being requisite for giving complete combustion with heavy oils are the mean temperatures during the whole process, and not the temperatures t_0 of the gas and air mixture immediately before ignition. The influence of the latter temperature has already been explained in the theoretical section.

The special problems of the oil turbine—on the one side, the maintenance of the temperature high enough to ensure combustion of the injected oil, and on the other side, the minimising of the heat loss to the walls—must therefore be solved by keeping the temperature of the combustion chamber walls at its most suitable minimum level,

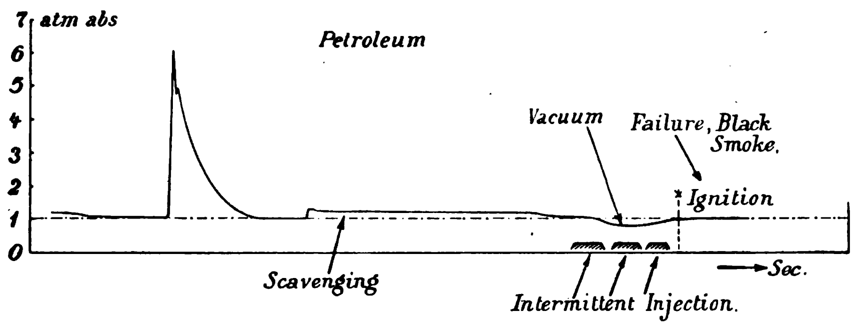


FIG. 93.—Diagram showing production of vacuum.

and at the same time providing large nozzle passages which can be closed.

Various liquid fuels were tested in the same way, ranging from benzine to tar oil. The fuel is pumped by a small pump to the ordinary type of pulverising valve, and air at a moderate pressure, supplied from a cylinder, serves to inject it.

It seems that the most suitable method is to inject the fuel intermittently, so that there is an air cushion between the individual spray injection.

Fig. 93, a pressure-time diagram taken from a test with petroleum, shows graphically the peculiar action which may take place in such cycles.

Following on normal explosion, there is a diminution of pressure (vacuum) in the combustion chamber; during the injection, so much heat is abstracted from the surrounding walls of the combustion chamber, that the pressure of the mixture in it is reduced to below atmospheric. In the ignition which follows, therefore, the combustion is very incomplete. In this case it is evident that the quantity of injected fuel, the injection pressure, and the chamber temperature were not in their proper relationship.

Following Sabathé's method, a combined test was made; after ignition and partial combustion, more fuel was injected into the mixture, and the time-pressure diagram shown in fig. 94 was obtained, but this arrangement was not successful. The combustion was less complete and the exhaust was smoky, and it may be mentioned that the heat loss by radiation to the walls was considerably increased.

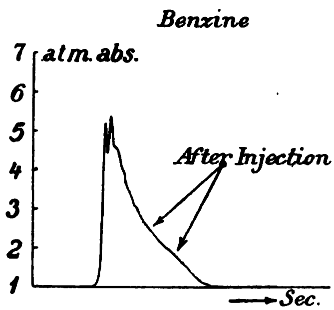


Fig. 94.

This combination trial was made as it had not been assumed that heavy oil could be employed with simple combustion.

It was found, however, that this could be done satisfactorily, and hence the necessity of the combined combustion and constant pressure method was avoided.

All the following experiments were carried out with simple combustion and intermittent injection of the fuel.

Figures relative to a few of the experiments are given below, together with the more important diagrams.

Heavy Petrol.

Date of test: 15th April 1910. Sp. gr. at 18° C. = 0·746.
Net calorific value about 10,000 calories per kg.
Price: 15.60 marks per 100 kg. Energy per H.P. = 645 calories.

TABLE VIII.

Time.	Mean Temperature in the Combustion Chamber.	Stroke of Pump.	Press. in Cylinder.	R.P.M. of the Cam Shaft.	Fuel per Charge.	Appearance of Exhaust.	Diagram.
	°C.	mm.	at. abs.		gr.		fig.
2·55	326	18	9·0	31	3·67	Heavy smoke	95
3·00	338	...	...	30	...	Scarcely visible	...
3·03	340	...	...	30	3·20	,, ,,	...
3·09	336	...	...	27	2·40	Colourless	96
3·21	351	...	...	30·5	2·94	,,	...
3·27	376	...	...	32	3·29	,,	...
3·58	388	15	9·3	30	...	,,	...
4·01	378	...	9·2	30·5	2·82	,,	97

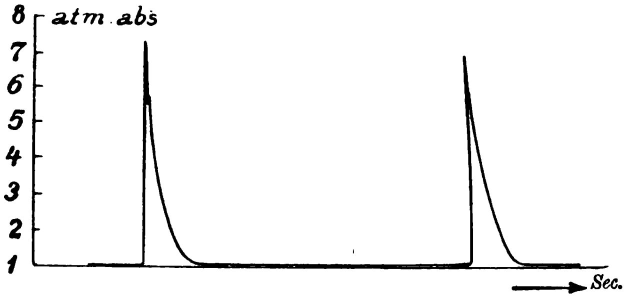


FIG. 95.

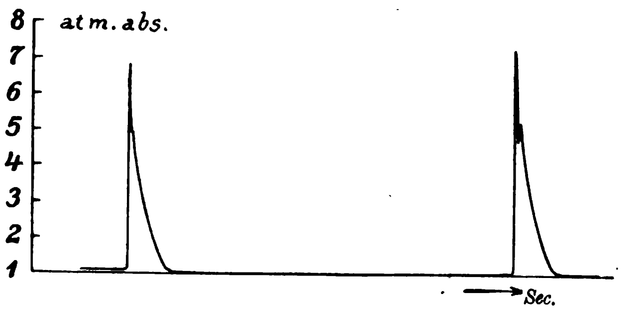


FIG. 96.

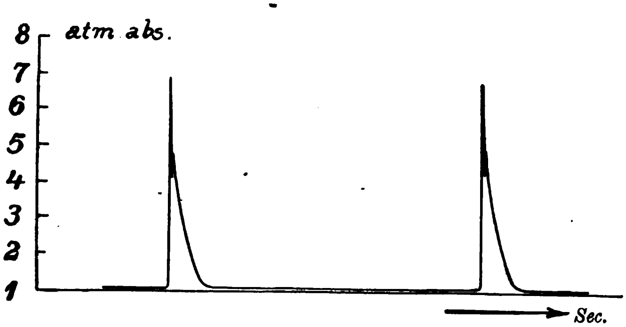


FIG. 97.

Heavy Motor Petrol II.

Date of test: 15th April 1910. Sp. gr. at 18° C. = 0·754.

Net calorific value about 10,000 calories per kg.

Price: 15 marks per 100 kg. Energy per H.P. = 668 calories.

TABLE IX.

Time.	Mean Temperature in the Combustion Chamber.	Stroke of Pump.	Press. in Cylinder.	R.P.M. of the Cam Shaft.	Fuel per Charge.	Appearance of Exhaust.	Diagram.
	°C.	mm.	at. abs.		gr.		fig.
4·35	361	15	7·3	29·4	4·18	Colourless	98
4·43	377	...	7·4	30	4·09	,,	...
5·51	374	...	7·8	29	3·72	,,	...
5·03	361	18	9·2	27	...	,,	99
5·07	367	...	9·2	26·5	4·35	,,	...
5·19	390	...	9·2	...	...	Rather smoky	100
5·27	379·5	...	10·7	29	3·02	Colourless	101

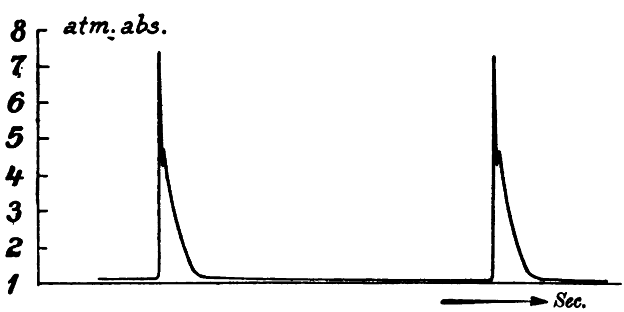


FIG. 98.

FIG. 99.

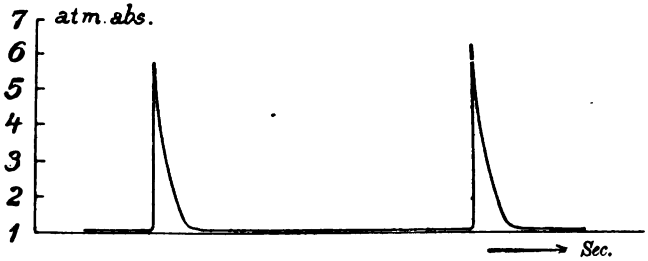


FIG. 100.

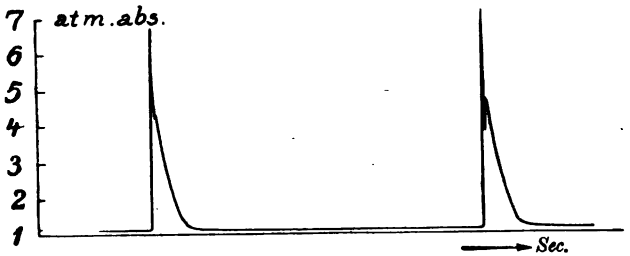


FIG. 101.

Petroleum.

Date of test: 12th April 1910. Sp. gr. about 0·8.

Net calorific value 10,300 calories.

Price: 12.30 marks per 100 kg. Energy per H.P. = 672 calories.

TABLE X.

Time.	Mean Temperature in the Combustion Chamber.	Stroke of Pump.	Press. in Cylinder.	R.P.M. of the Cam Shaft.	Fuel per Charge.	Appearance of Exhaust.	Diagram.
	° C.	mm.	at. abs.		gr.		fig.
4·56	412	15	9·9	33	3·56	Slightly bluish	102
5·02	423	...	...	...	...	,, ,,	...
5·03	422	...	...	30·7	3·9	,, ,,	...
5·23	370	12	7·5	29·9	...	Very slightly blue	...
5·30	382	...	7·6	30	...	Nearly colourless	103
5·35	382	...	7·5	29	3·06	,, ,,	...
5·46	412	...	6·8	30	2·91	,, ,,	...
5·56	413	...	6·8	30	3·30	,, ,,	104

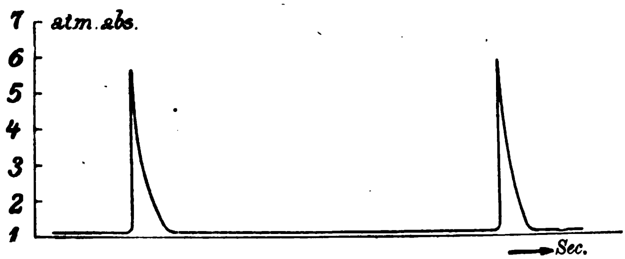


FIG. 102.

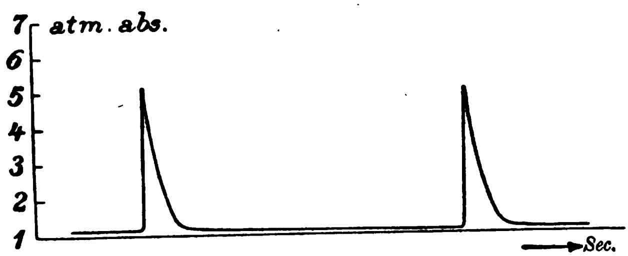


FIG. 103.

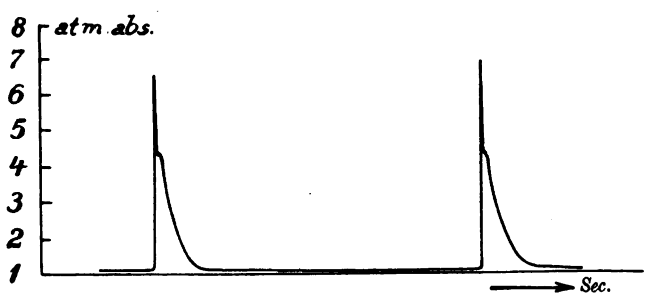


FIG. 104.

German Gas Oil from Schulau (Holstein).

Date of test: 22nd April 1910. Sp. gr. 0·85.

Net calorific value about 10,800 calories.

Price: 10.52 marks per 100 kg. Energy per H.P. = 1030 calories.

TABLE XI.

Time.	Mean Temperature in the Combustion Chamber.	Stroke of Pump.	Press. in Cylinder.	R.P.M. of the Cam Shaft.	Fuel per Charge.	Appearance of Exhaust.	Diagram.
	°C.	mm.	at. abs.		gr.		fig.
4·50	438	12	9	29·8	3·36	Practically colourless	105
5·00	439	...	9	28·3	...	...	106
5·11	436	...	8·5	29·4	...	Slightly bluish	107
			Roumanian Gas Oil.				
5·40	452	9	9	26·8	3·56	Slightly visible	108
5·52	447	9	9	26·8	3·56	Practically colourless	109

Date of test: 9th May 1910. Sp. gr. 0·845.

Net calorific value about 10,200 calories.

Price: 8.60 marks per 100 kg. Energy per H.P = 1180 calories.

FIG. 105.

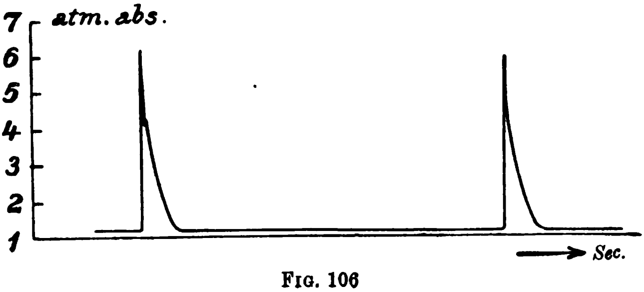


FIG. 106

Fig. 107.

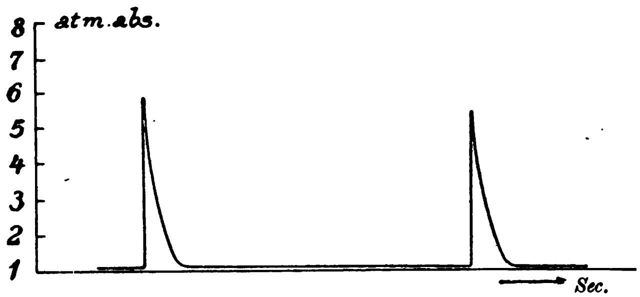


Fig. 108.

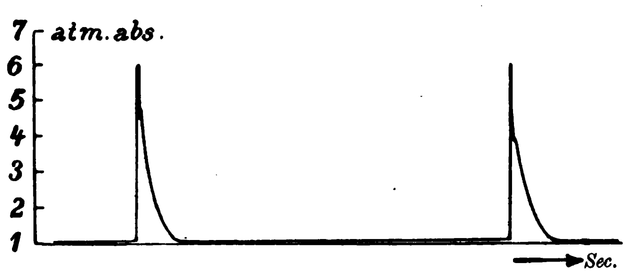


Fig. 109.

Benzol.

Date of test: 18th April 1910. Sp. gr. 0·875.

Net calorific value about 9700 calories.

Price: 18.00 marks per 100 kg. Energy per H.P. = 540 calories.

Mean fuel consumption per charge = 3·92 gr.

TABLE XII.

Time.	Mean Temperature in Chamber.	Stroke of Pump.	Press. in Cylinder.	R.P.M. of the Cam Shaft.	Exhaust.	Diagram.
	°C.	mm.	at. abs.			fig.
3·13	295	16·5	9·0	30	Slightly smoky	110
3·18	326	...	...	30·6	,, ,,	...
3·23	344	...	...	29·6	Very slightly smoky	...
3·27	353	...	...	30·2	Still less smoky	...
3·32	359	...	...	30·2	Colourless	111
3·37	358	...	...	29·8	...	...
3·43	354	...	...	29·8	...	...
3·47	387	...	...	30·5	...	...
3·51	396	...	...	29·0	...	...
3·56	401	...	...	30·5	...	...
4·01	379	...	10·0	30·2	...	112
4·06	352	...	10·2	29·8	...	...
4·19	378	...	6·0	26·8	...	113
4·23	382	...	6·0	27·1	...	...

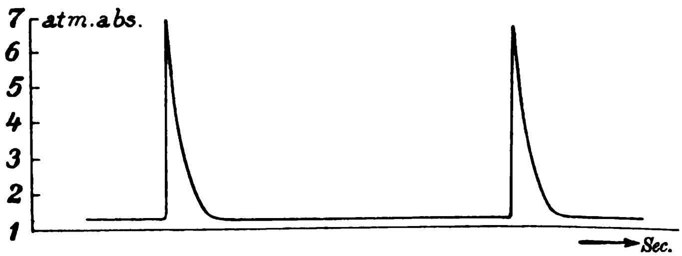


FIG. 110.

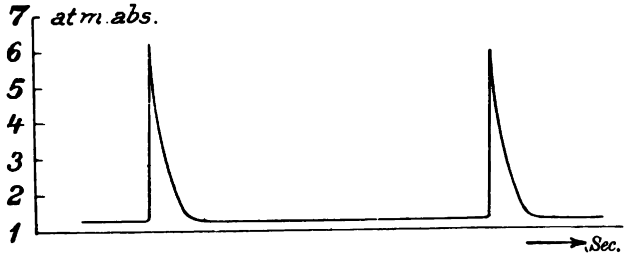


FIG. 111.

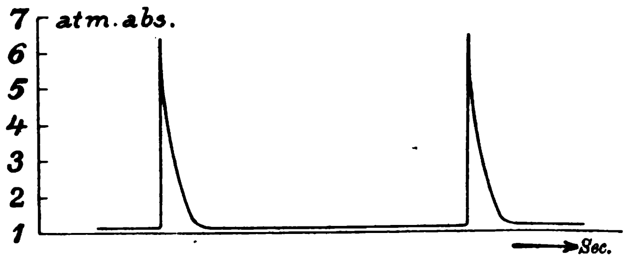


FIG. 112.

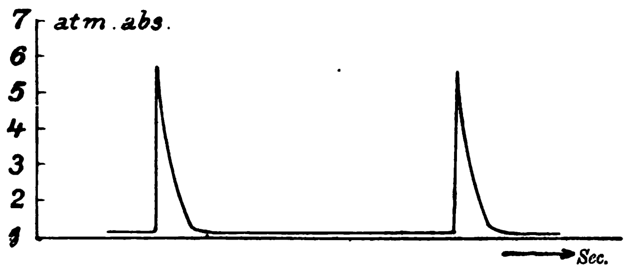


FIG. 113.

Tar Oil.

Date of test: 21st May 1910. Sp. gr. 1·05.

Net calorific value about 9300 calories.

Price: 4.00 marks per 100 kg. Energy per H.P. = 2320 calories.

Mean fuel consumption per charge = 2·77 gr.

TABLE XIII.

Time.	Mean Temperature in Chamber.	Stroke of Pump.	Press. in Cylinder.	R.P.M. of the Cam Shaft.	Exhaust.	Diagram.
	°C.	mm.	at. abs.			fig.
3·07	386	12	8·5	29·5	Colourless	114
3·16	380	...	...	27·5	...	115
3·20	379	...	...	28·4	...	116
3·26	403	...	...	31	...	117
3·36	425	13	...	30	Smoke formed from vacuum	...
3·39	373	12	...	28·2	Colourless	118

In all these experiments the capacity of the chamber was 50 litres.

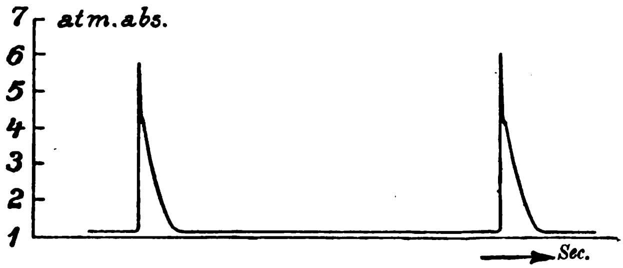


FIG. 114.

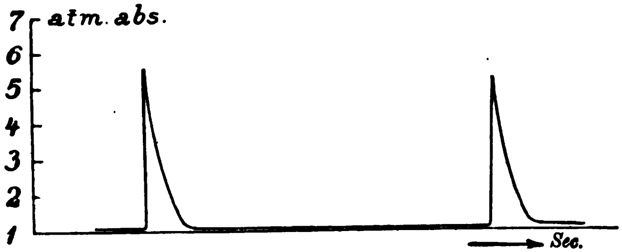


FIG. 115.

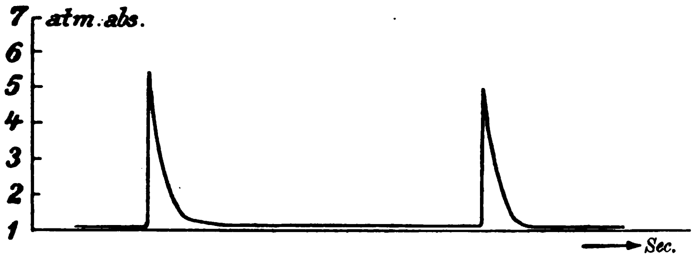


FIG. 116.

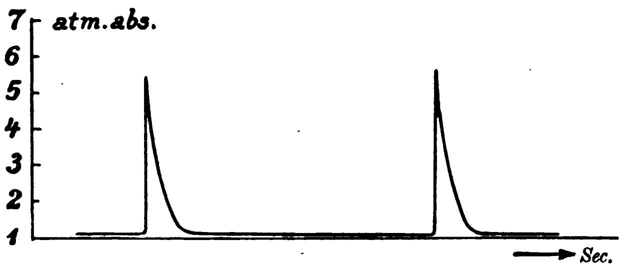


FIG. 117.

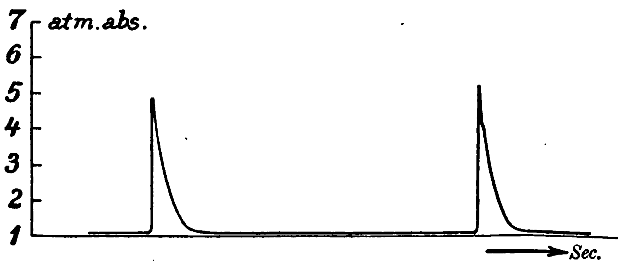


FIG. 118.

2. EXPERIMENTS ON THE FIRST TURBINE USED FOR DRIVING DYNAMO.

Fig. 119 shows the general arrangement of the first gas turbine plant for dynamo driving, including the power gas plant and the regenerator.

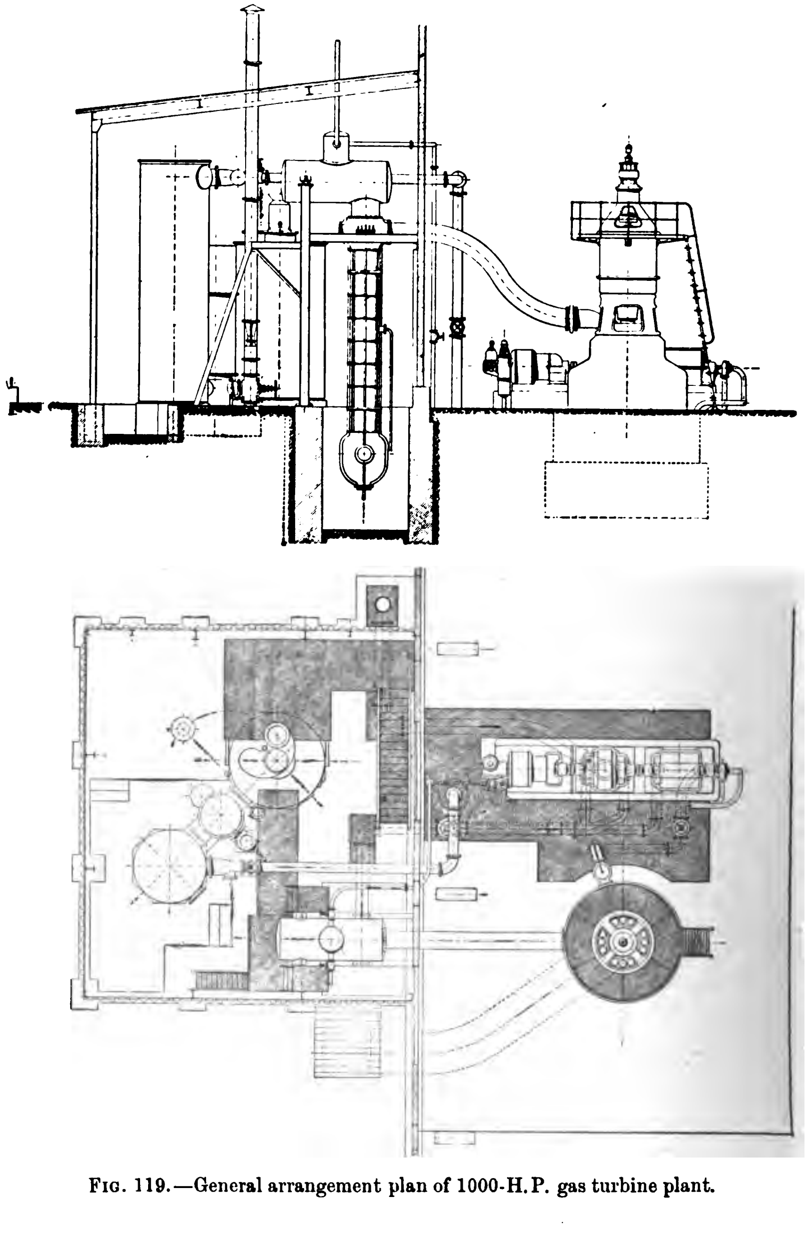


FIG. 119.—General arrangement plan of 1000-H.P. gas turbine plant.

Figs. 120, 121, and 122 are photographs of the actual plant, taken from various points.

Fig. 120.—First turbine for driving purposes, seen from above.

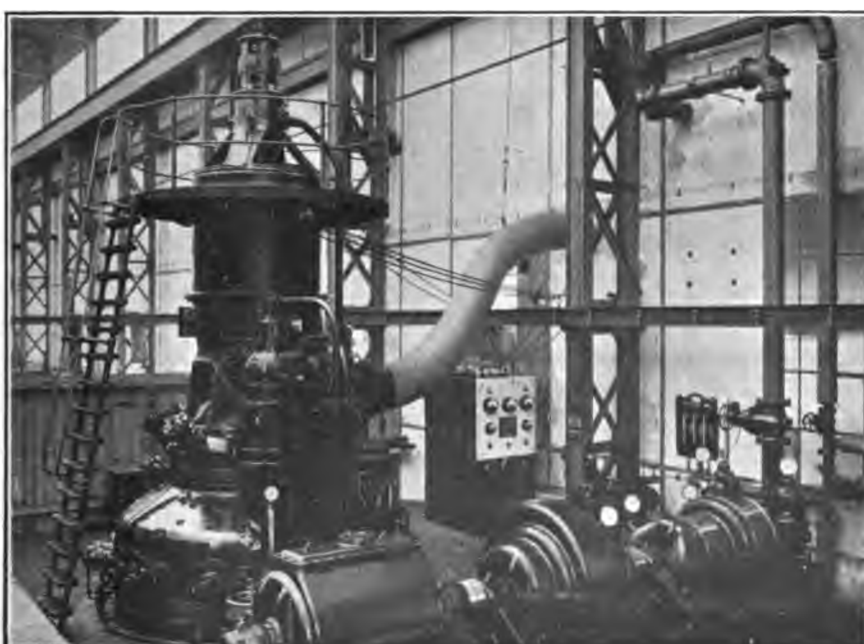


Fig. 121.—Side view.

The design of this plant was commenced in the early summer of 1909, and the erection was completed in the winter of 1910.

The plant was arranged for operation under normal working conditions. Above all, it was desired to test the efficiency of this new prime mover, since the trials on the original turbine had shown that the process could be satisfactorily carried out without any mechanical troubles arising.

Whilst the fundamental points were very quickly settled by tests on the original turbine, the construction of an efficient engine involved some

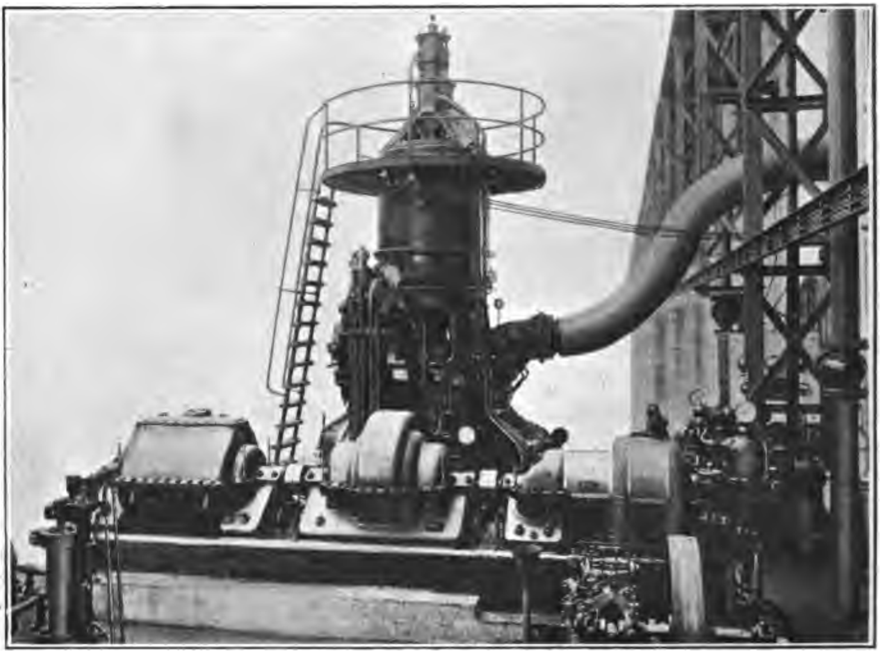


FIG. 122.—View facing blowing plant.

unexpected difficulties. As long as all the basic physical changes and actions involved were not clearly understood—and this was the case at first, as the arrangement was quite novel—the process could not reach the degree of completeness which was necessary for an efficient machine. It was essential to strive at a determination of what was actually attained, and to completely neglect all the accidental effects in the operation which might arise.

The difficulties encountered were not of a mechanical nature, but were almost entirely thermo-dynamic ones.

The results obtained had to serve to build up the gradual "Theory of the Gas Turbine," and only a theory so obtained is of value. In the tests begun in this year, power gas from our own gas plant was exclusively used. The gas blower drew the gas from the scrubber and delivered it to the gas turbine under pressure.

Later, anthracite was employed for gas-production, in order to provide gas rich in hydrogen and hydrocarbons. The large percentage of these constituents is valuable in the combustion process, owing to the higher ignition velocity of hydrogen (35 m. per sec.) as compared with carbon monoxide (2·2 m. per sec.).

Unfortunately, the turbine blower running at 5000 r.p.m. acted as a tar separator, and the employment of anthracite with a turbo-blower was found to be unsuitable. Coke used in metallurgical work was then exclusively utilised, and the gas generated was the poorest in hydrogen and heating value that was used in the gas turbine tests. 12 to 16 per cent. of hydrogen was the highest value that could be reached, and there were no hydrocarbons present. Power gas from anthracite gives 23 per cent. hydrogen and a certain amount of hydrocarbons. This circumstance rendered the experiments more difficult to a not inconsiderable extent, but it had the effect of prolonging the trials until complete combustion was obtained with gas containing a small proportion of hydrogen.

It is interesting to note that the explosion process in the reciprocating gas engine, owing to pre-ignition, is more difficult or less economical with gas rich in hydrogen and hydrocarbons than with a gas containing a large percentage of CO; and, on the other hand, with the gas turbine it is preferable for the gas to contain a large proportion of hydrogen.

The "Theory of the Gas Turbine," given earlier in this volume, forms in itself a general summary of the tests. The following detailed points regarding the tests may be mentioned.

In spite of a most careful and efficient arrangement of the oil-operated control gear, it was found that the combustion process was unstable, although this method of control has proved satisfactory and preferable to the mechanical control, and even appears absolutely necessary for a practical gas turbine.

In February 1911 the turbine was run on no load, with full excitation 4000 volts, at 3000 r.p.m., consuming only 250–300 cub. metres of gas per hour with a heat content of 1179 calories. This corresponded to an efficiency of over 20 per cent. (if the air resistance of the disc be taken with that of the dynamo rotor). This was certainly a satisfactory starting-point for the efficiency curve.

Unfortunately, this was only transitory, and strenuous attempts had to be made to render the working more stable.

The arrangements for the air supply were then modified. As can be seen from fig. 66, the air was drawn in through the turbine shaft from the atmosphere, and the turbine then runs noiselessly. Owing to contact with the walls of the chambers, however, the air becomes considerably heated.

On this account the air was taken direct from the engine-room. The side openings of the air valve were closed up, and they were bored on the front surface. The air thus entered direct from the engine-room through the air valves into the combustion chamber.

Further, the air can pass round the blade passages, which were designed for high speed, so that a small drop is sufficient to force the air through the nozzles into the turbine casing.

After this alteration the operation was much more even and regular. Tests of some hours' duration were able to be made, and temperature measurements taken at various portions of the combustion chamber and in the exhaust. These measurements provided very important data, which were utilised in the earlier sections of this volume, namely, (1) those on the influence of the temperature t_0 of the mixture before ignition; (2) influence of the size of the combustion chamber; (3) explosion wave.

The calculations which were made in these sections are based on the tests mentioned above, and they may be taken equally well in connection with this chapter as in the sections stated.

Fig. 55 shows at sight the efficiency which can be and should be reached with this machine.

Figuratively speaking, the following process was adopted in these experiments:—At the commencement of the test the value for the efficiency was obtained (with a small charge) which corresponded with the ratios in fig. 55. By gradual heating we descended, figuratively speaking, from the efficient summit. Finally an exhaust temperature of 400° C. was reached, and it was not desired to exceed this.

The chambers were then well cooled by cooling water in addition to the cooling effect of the scavenge air, and the ignition was improved. Previously the explosion wave prevented good ignition, which was in fact seemingly deficient.

By this means the temperature t_0 before ignition was considerably reduced, and the process became completely stable. The improvement in the ignition had the effect, moreover, that the ignition and combustion were now as effective as could have been anticipated from the theoretical considerations in the section dealing with this point in the "Theory of the Gas Turbine," and the process did not give rise to an explosion wave.

This was the case not only for small charges, but also for the maximum charges, with which the combustion had always previously been comparatively incomplete.

Thus a stage was reached in the development of the gas turbine which is attained in the reciprocating gas engine when the indicator diagram is the best possible one which can be obtained, and remains constant.

In the following experiments it was endeavoured to follow out and build up the working cycle in the turbine, so that the power actually developed might be equal to that calculated. It was then found that,

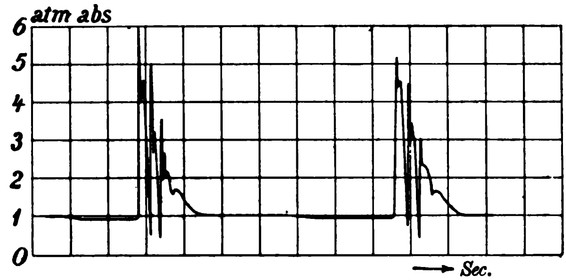


FIG. 123.—Diagram taken behind the nozzle valve.

with a large charge of gas, the pressure in front of the nozzle valve decreased continuously and regularly during expansion, whilst behind the nozzle valve the diagram had a very irregular appearance (fig. 123).

This undesirable effect was due to the fact that, after the opening of the nozzle valve through the explosion, it closed repeatedly owing to the

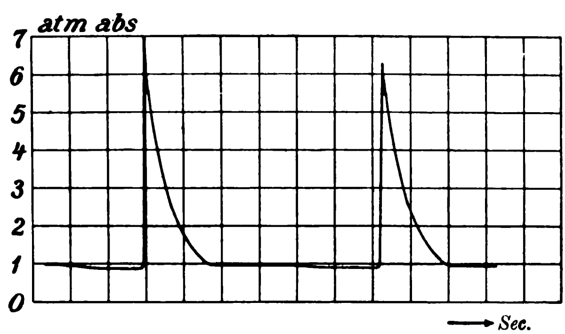


FIG. 124.—Diagram taken behind the nozzle valve.

oscillation in the regulating oil-pressure column. When the regulating piston of the nozzle was no longer arranged to have a constrained motion in order to make the first closing of the nozzle valve extremely rapid, this action, with its bad effect on the turbine process, was removed.

The next step was a determination of the power curve with increasing charge of gas, and therefore increasing explosion pressure. The foregoing tests gave an indication of the general character of this curve, and its determination for several points was made from 12th to 14th October 1911 without any trouble.

Fig. 125 shows the manner in which the maximum explosion pressure depends on the weight of gas supplied per charge; the p_1 line represents the theoretically attainable values for $t_0 = 15°$ C. and $t_0 = 100°$ C. The points marked are those obtained by experiment. The numbers marked in figs. 125 and 126 refer to the indicator diagrams, figs. 127–136, which were taken in front of the nozzle valve.

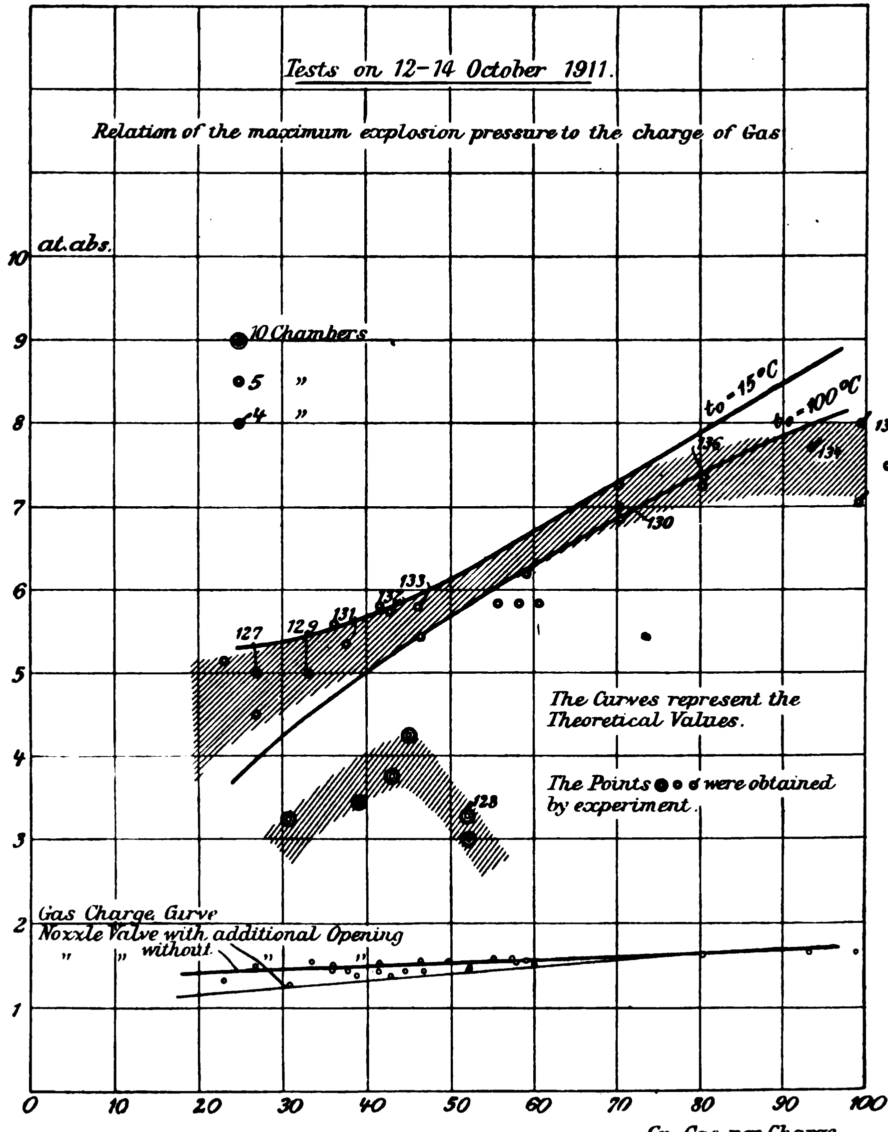


FIG. 125. p_1—G curves.

Fig. 126 shows the relation of the power developed per chamber to the maximum explosion pressure p_1. As mentioned above, the numbers of the indicator diagrams given below are marked beside the corresponding points obtained in the experiments.

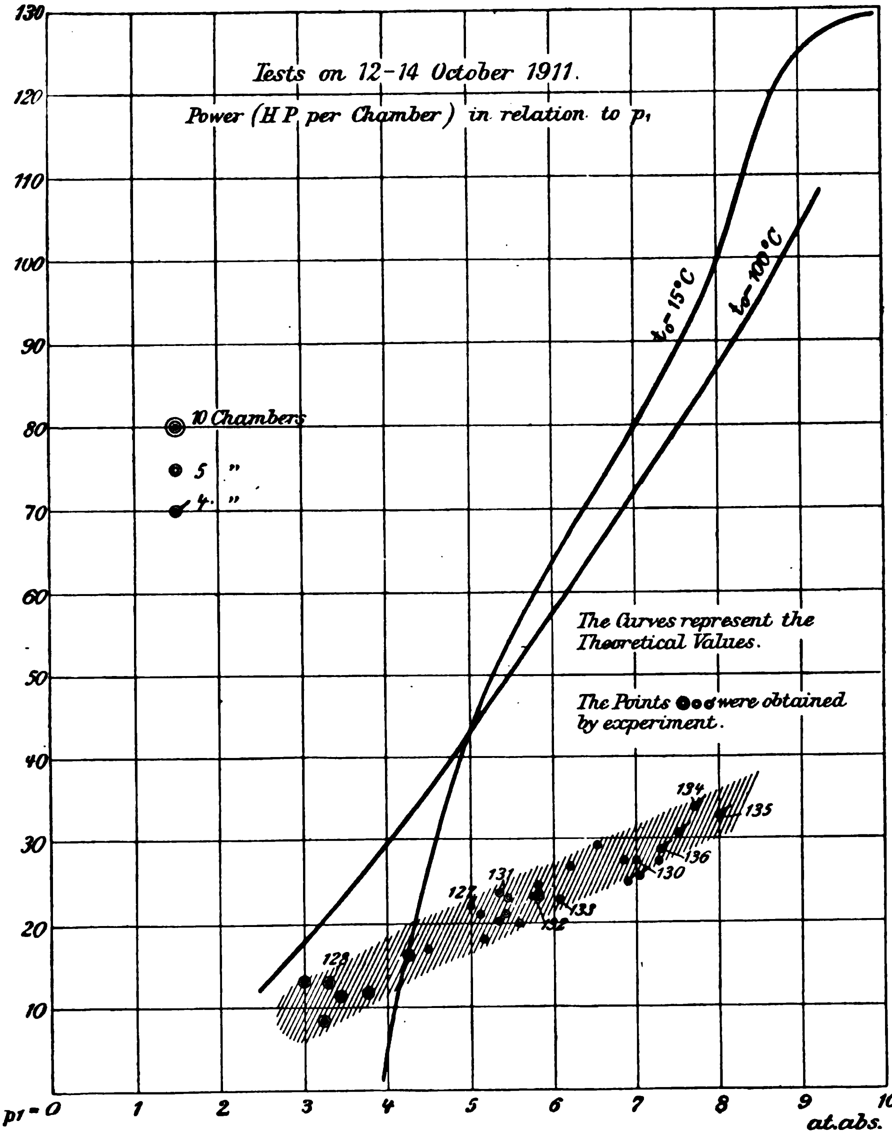


FIG. 126. N—p_1 curves.

From fig. 125 experimental confirmation is obtained of the fact that combustion with four or five chambers in operation is as complete as is possible. The values obtained by experiment lie between the two curves for complete combustion obtained by theoretical calculation, when $t_0 = 15°$ C. and $t_0 = 100°$ C., the heat content of the gas being 1179 calories.

It is only when the gas charge exceeds 80 gr. per charge that deviations of a material nature begin to appear. Small variations occur with gases of different constituents, particularly where there is a difference in the percentage of hydrogen.

There is a considerable change, however, when ten chambers were in operation. In this case the values of p_1 were below those with five chambers. Above 50 gr. per charge the process could not be carried out with ten chambers, as the emergency governor came into action.

Fig. 126 shows that, with explosion up to just above 4 atm. abs., the power developed per chamber was between the calculated values obtained with $t_0 = 15°$ C. and $t_0 = 100°$ C.

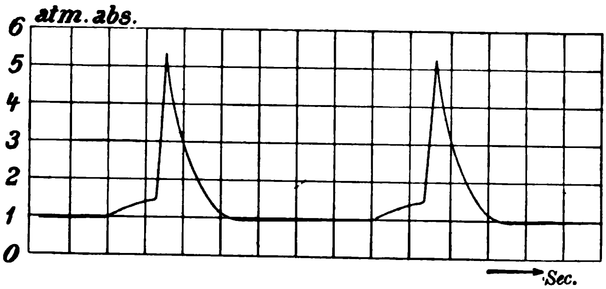


Fig. 127.

In this case, as previously mentioned, the resistance of the disc was not separated from that of the rotor of the dynamo, as this could not be done experimentally.

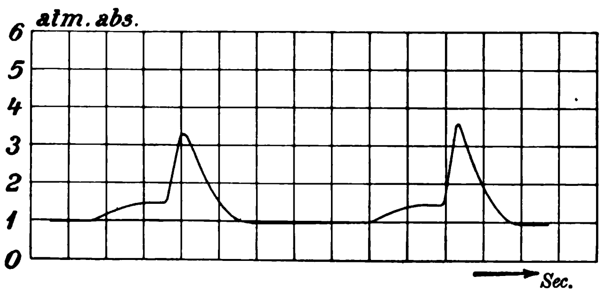


Fig. 128.

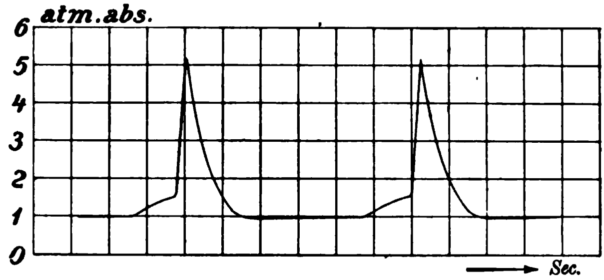


FIG. 129.

Above $p_1 = 4{\cdot}2$ atm. abs. the average experimental values did not coincide with those obtained by calculation, the power being less than it should be according to calculations.

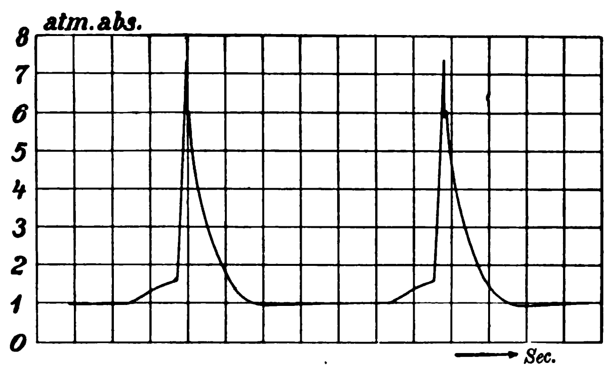


FIG. 130.

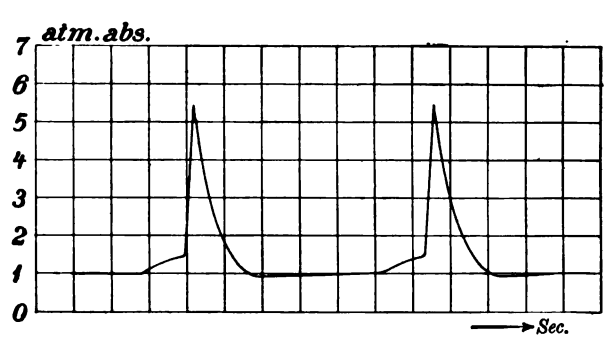


FIG. 131.

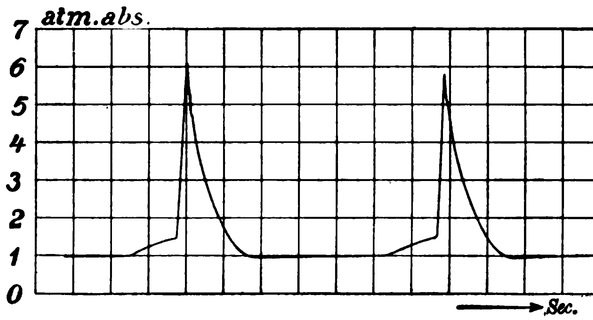


FIG. 132.

Moreover, when p_1 was greater than 7 atm. abs., the temperature t_0 was greater then 100° C., which is undesirable. These points must therefore not be compared with the line $t_0 = 100°$ C., but with a line $t_0 > 100°$ C. below it.

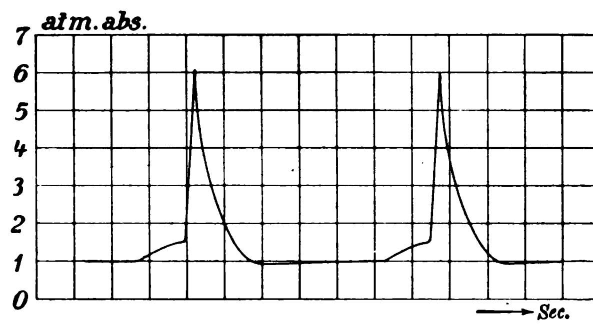


FIG. 133.

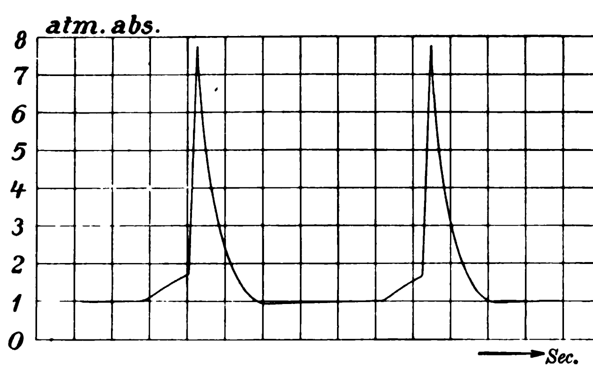


FIG. 134.

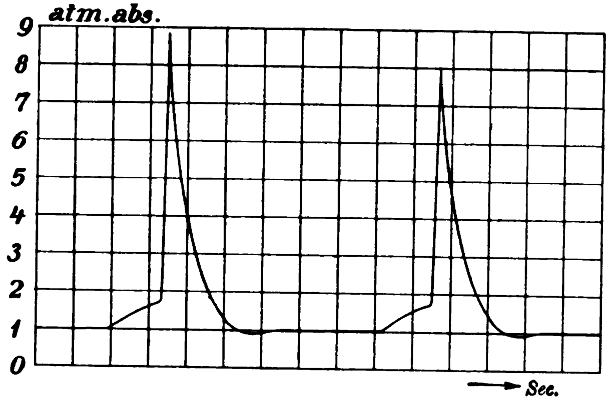


FIG. 135.

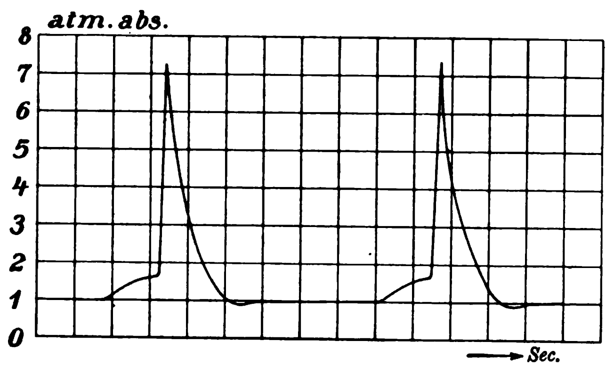


FIG. 136.

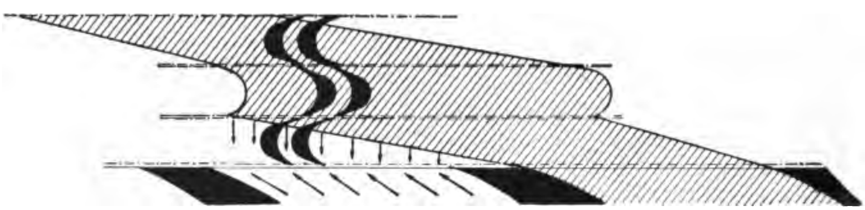
Two further observations may be made. The vacuum pressure gauge on the exhaust pipe, about 1500 mm. from the opposite nozzle, indicated momentary pressure rises of about 0·8 m. water immediately after a big discharge from the orifice. Some very powerful sound vibrations evidently arise.

Finally, measurements were made of the clearance pressure at the outlet of the nozzle, and it was found that with $p_1 = 3$ atm. abs. the pressure of the stream in the clearance space was about 1 m. water pressure higher than the pressure surrounding the clearance space.

It should in conclusion be noticed that, as shown in fig. 137, the discharged stream of exhaust gases moving with a high velocity passes nearly directly over the nozzle of the adjacent chamber, from which at the same time scavenge air is flowing out if ten chambers are in operation.

It is clear, therefore, that the observations as stated above depend upon the same cause. The nozzles, with the conditions as in the tests on 12th to 14th October, were suitable as regards the pressure drop only when the value of p_1 was less than 4·2 atm. abs. Above that figure the velocity of flow was only slightly increased, in spite of the rise in pressure at the exit from the nozzle; the flow of gas is not broken up by the vacuum from the exhauster, and gives rise to very strong sound vibration as it leaves the nozzle. These pressure waves are disadvantageous in preventing the necessary scavenging of the chambers, if ten are in operation. Too great expansion in the nozzles helps the scavenging of the adjacent chamber, owing to the suction of the stream of gas; insufficient expansion, as above, hinders the scavenging of the adjacent chamber to a large extent.

When five chambers are in operation, on the other hand, the nozzle of the chamber which is being scavenged does not lie under the stream of gas from the next working chamber, since between the two there is a chamber out of action.



Scavenge air. *Exhaust gas.*

FIG. 137.—Impeded scavenging.

In steam turbines, the disadvantage of insufficient expansion in the nozzles cannot be counteracted, as the clearance between the nozzles and the first moving wheel must be small; expansion then takes place in the moving wheel, and almost the whole of the energy thus set free is used in accelerating the flow of steam.

It is different in the gas turbine, where, for the purpose of a more convenient and simpler arrangement for the air supply, there is a comparatively large clearance between the nozzles and the moving wheel, and moreover the clearance space must be in communication with the space behind the last moving wheel, owing to the peripheral passages. In this case the expansion of the gas after leaving the nozzles brings it to the pressure in the exhauster, and further acceleration does not occur.

The degree of expansion in a conical nozzle can be determined by the nozzle expansion, *i.e.* the ratio of the section at the exit to the smallest section. For the calculation of this ratio the theory of the steam turbine provides certain formulæ which depend on the pressure ratio, and these formulæ are used for the determination of the nozzle expansion. If, however, calculations be made following on the procedure adopted on page 16, in which the pressure and temperature fall are introduced (taken from the entropy diagram), considerably higher values are obtained for the nozzle ratio—nearly double those in the previous case.

Comparing the ratios obtained by this new procedure with the nozzle ratios which were found by the experiments to be much too small, and making due allowance, very good agreement is obtained. This indicates that the new method of calculation of the nozzle ratios for the gas turbine is accurate. With the employment of new nozzles, the power curve must coincide with the one previously calculated, even where the explosion pressure is higher than 4·2 atm. abs.

This stage in the experiments adds further information for the "Theory of the Gas Turbine."

It should be mentioned that the examples given in the theoretical portion are taken from experiments on both the turbines. They do not, therefore, indicate the maximum values which can be obtained from the newest type of engine, but the highest which have been obtained from the machines tested. In particular, the efficiency curves fig. 55, etc., are capable of being increased by further development of the gas turbine by having larger nozzle outlets, by raising the pressure of the charge, and also the maximum explosion pressure, by substitution of the nozzle valves by flap valves, and allowing unrestricted flow for the gases, etc.

Finally, it may be repeated that there were no mechanical troubles of any sort, and no corrosion or rusting, in spite of the fact that the tests have been proceeding for three years. This is no doubt entirely due to the fact that no water in any form is used in the process. The vapour which is produced by the combustion remains so highly superheated that it has no deleterious effect on any portion of the plant.

It may be remarked that troubles arising from the formation of acid very rapidly show themselves, and after fitting a cooling jacket a few chambers only showed a slight effect.

The electrodes of these chambers showed very severe corrosion after a few days. The wrought-iron hollow section of the electrode had ½ mm. of rust, and even the nickel-steel electrode was corroded. It was not pure rust, but probably sulphate or nitrate.

This effect took place in spite of the fact that the coke used was exceptionally free from sulphur. The unavoidable constituents of sulphuric and nitric acid in the presence of a small quantity of water must therefore have a very rapid destructive action.

Air scavenging renders the presence of water during the process unnecessary. That alone guarantees the life of the gas turbine.

APPENDIX.*

In the review of the German edition of this volume, Dr Stodola† entered particularly into the question of the efficiency of the explosion gas turbine in which the pressure of the charge is relatively low.

Further tests which have been carried out by the author at a more recent date are interesting as showing that the results given previously by no means represent the maximum limits which may be reached by the explosion gas turbine.

In Table I. are given the calculated values for the theoretical process for varying charges of gas and also for varying initial temperatures. These values naturally cannot be taken as applying generally, but are true only for the plant which was tested. The calculations are made with reference to the specific heat and the heat losses, and serve to express the actual relative values. For the combustion process the general analytical method is employed; for the expansion process, Stodola's gas entropy diagram is employed, with the graphical method, which alone makes allowance for the heat losses and the variations of the specific heat.

The calculations thus do not differ at all in principle from those employed previously in the book; there are certain modifications of detail, such as the introductions of the values of p_0 for varying values of t_0, and of the time of expansion with variation of p_1, the calculations for values of t_0 equal to 150°, 200°, and 300° C. being added.

As regards the turbine efficiency, all the calculations in this and the following tables are based on those in fig. 43.

In Table II. the same calculations are made, also with reference to the gas entropy temperature diagram, but the assumption is made (which is never quite the case) that there is no heat loss. In these calculations allowance is made for the variation of specific heat.

In Table III. the same procedure is adopted, based on Stodola's deductions from his analytically derived formula. According to this formula, Stodola comes to the conclusion that the expression $\frac{p_2}{427}(v_2 - v_1)$ must be subtracted from $\int c_v dT$. $\int c_v dT$ is again derived from the entropy table, which therefore takes into account the variation in specific heat, and v_2 and v_1 are also obtained from this table. It is assumed, as in Table II., that no heat loss occurs.

The main results of the various calculations are shown in figs. 1, 2, and 3. In fig. 1 the curves indicate the power of the turbine wheel in relation to the

* Translated from an article in the *Z. d. V. d. I.*

† *Z. d. V. d. I.*, p. 527, 1911.

quantity of gas supplied, in cubic metres per hour. The heavy-line curves represent the theoretical values from Table I., and the values are thus based on the author's previous method of calculation. The thinner curves above, represent the values from Table II.; and if there be no heat loss, these curves would, in the author's opinion, be obtained, having reference to the turbine efficiencies indicated in fig. 43. The thinner curves below correspond to Table

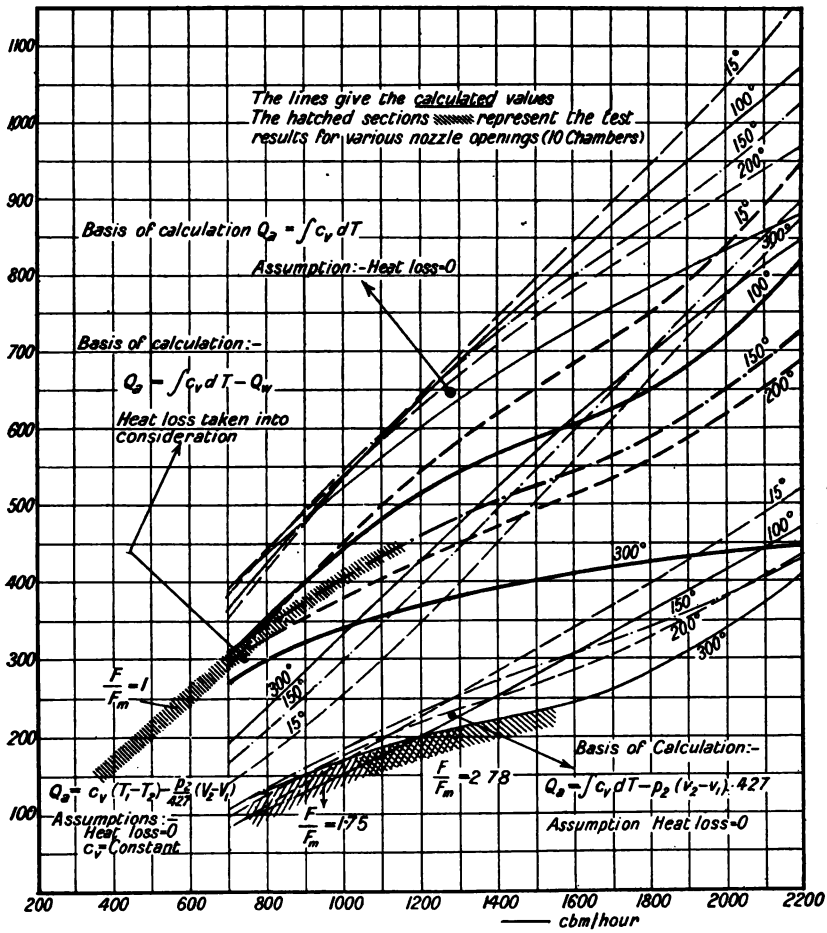


FIG. 1.—H.P. - *cbm* curves for $t_0 = 15°$, 100°, 150°, 200°, 300° C.

III., that is, to Stodola's general formulæ. In contrast to these theoretical values, the hatched curves give the actual values obtained with the gas turbine.

With $\frac{F}{F_m} = 1{\cdot}75$ the hatched curves modify the results obtained in October 1911. The wattmeter originally employed was inaccurate, as it had not been previously calibrated.

In these tests, nozzles were employed giving the ratio $\frac{F}{F_m} = 1{\cdot}75$. The symbols are the same as in the other tests. In the tests with $\frac{F}{F_m} = 1$ and $\frac{F}{F_m} = 1{\cdot}75$, the section at the exit (F) remained the same, whilst in the tests $\frac{F}{F_m} = 1{\cdot}75$ and $\frac{F}{F_m} = 2{\cdot}78$, F_m was constant.

In a similar manner, in fig. 2 the efficiency is shown in relation to the weight

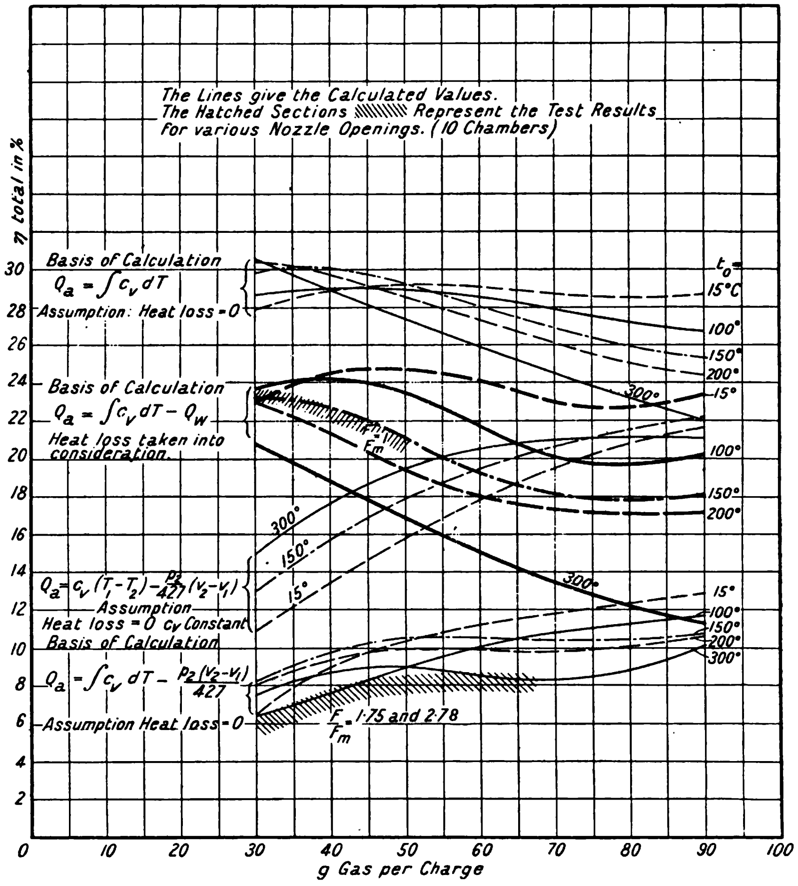


FIG. 2.—Efficiencies for $t_0 = 15°, 100°, 150°, 200°, 300°$ C.

of gas supplied per charge, and in fig. 3 the power (in H.P.) per chamber in relation to the explosion pressure p_1. Only the results of the tests with ten chambers were inserted, as those with five chambers remain as before. Apart from the modification in the nozzles, and the other variations thereby occasioned, there were no alterations in the machine.

Whilst figs. 1 and 2 only give the final results of the combustion and expansion processes, fig. 3 gives the complete results for the expansion process, These three sets of graphical representations of the results of the tests and the values obtained by calculation lead to the following conclusions.

The best test results (with $\frac{F}{F_m} = 1$) fall within the curves for the theoretically calculated values, following the author's method. The curves from Stodola's last general formula lie considerably below these test results for $\frac{F}{F_m} = 1$.

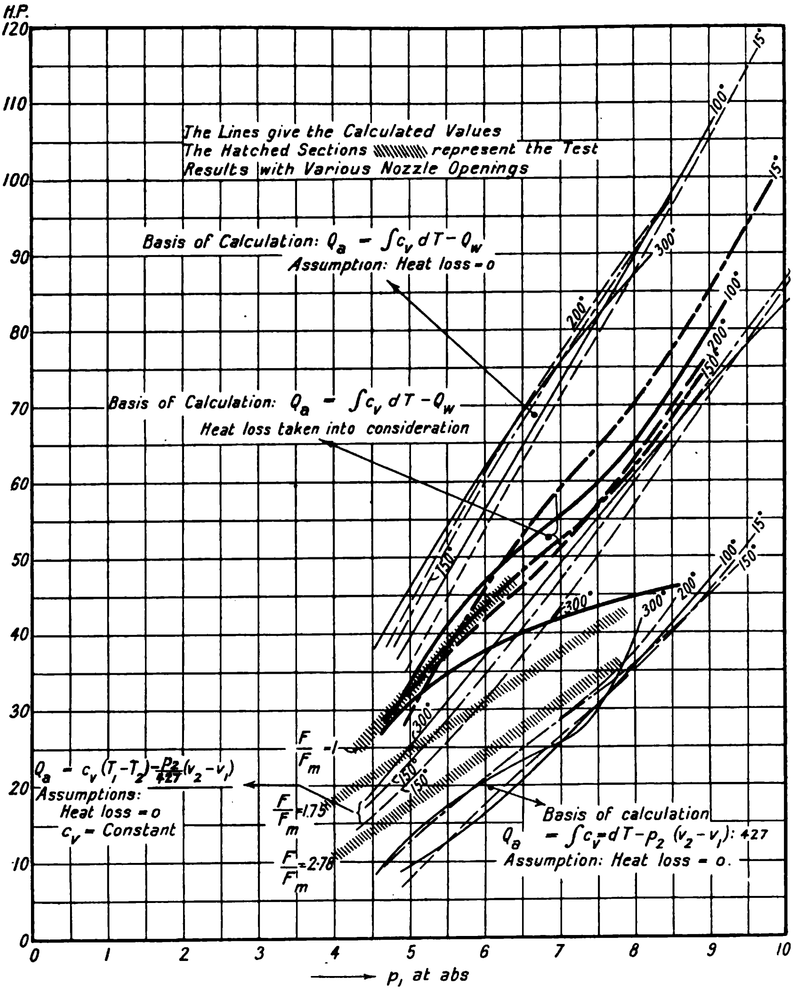


Fig. 3.—$N - p_1$ curves for $t_0 = 15°, 100°, 150°, 200°, 300°$ C.

It is only in the curves in figs. 1 and 2 that the test values for $\frac{F}{F_m} = 1{\cdot}75$ and $\frac{F}{F_m} = 2{\cdot}78$ lie within the curves of Stodola. Apparently, therefore, he has based his theory on this first test. Fig. 3, however, in which no account is taken of the combustion process, shows clearly that the results with the most unsuitable nozzles still lie above the curves deduced by Stodola.

The best test results in fig. 3 are about 100 to 200 per cent. above Stodola's curves, which presuppose no loss of heat to the walls.

It would therefore appear that the general formula on which Stodola based his criticism is incorrect, which is undeniably proved by the tests. Even from purely theoretical considerations, it can be proved that Stodola's formula is not correct, as a general formula. This was shown by Dr Mangold of Augsburg in January 1912, by means of analytical methods.

At the commencement of the flow of gas let the conditions be represented by $p_1 v_1 T_1$, the weight by G_1, and the volume of the space from which the gas passes by V_1. Let p, v, c, G represent the pressure, specific volume, final velocity, and weight of gas at a particular moment of its passage, and let $p_2 v_2 T_2$ represent the conditions at the end of expansion.

Then
$$L = \int_1^2 \frac{Ac^2}{2g} dG = \int_1^2 (i - i_2) dG$$
$$= \int_1^2 kc_v(T - T_2) dG \qquad (1)$$

L being in calories.

And
$$G = G_1 - \frac{V_1}{v}; \; dG = \frac{V_1}{v_2} dv \qquad (2)$$

$c_v dT = Apdv$ for adiabatic expansion,

$$v = v_1 \left(\frac{T_1}{T}\right)^{\frac{1}{k-1}},$$

$$dG = -\frac{1}{k-1} \cdot \frac{V_1}{v_1} \cdot dT \cdot \left(\frac{T}{T_1}\right)^{\frac{1}{k-1}},$$

$$L = G_1 \int_1^2 -\frac{k}{k-1} \cdot \frac{dT}{T} \cdot \left(\frac{T}{T_1}\right)^{\frac{1}{k-1}} c_v \cdot (T - T_2),$$

or as a general equation for adiabatic expansion,

$$L = G_1 \int_1^2 c_v(T - T_2) \cdot \frac{k}{k-1} \cdot \frac{1}{\left(\frac{T_1}{T}\right)^{\frac{1}{k-1}}} \cdot \frac{dT}{T} \qquad (3)$$

To follow this investigation further, let it be assumed that the specific heat is constant as well as that the expansion is adiabatic. Integrating equation (3) between the limits T_1 and T_x, we have (taking $G_1 = 1$ kg.)

$$L = c_v(T_1 - T_x) - c_v(kT_2 - T_x)\left(1 - \left(\frac{T_x}{T_1}\right)^{\frac{1}{k-1}}\right) \qquad (4)$$

This simplifies to the following equation in the special case $T_x = T_2$:

$$L = c_v\left[T_1 - T_2\left(k - (k-1)\left(\frac{T_2}{T_1}\right)^{\frac{1}{K-1}}\right)\right],$$

or by substitution

$$L = c_v(T_1 - T_2) - \frac{p_2}{427}(v_2 - v_1) \qquad (5)$$

Equation 5 of Mangold agrees with the last but one of Stodola, but both serve expressly only when c_v and c_p are constant, $Q_w = 0$, and integrated between the limits T_1 and T_2; therefore the equation is not suitable for intermediate values.

From equation (5), with the same assumptions as before, Table IV. is calculated, but for initial temperatures of $t_0 = 15°$ C., 150° C., 300° C. The final results are again shown by the thin lines in the figs. 1, 2, and 3. The curves lie considerably above those corresponding to Stodola's formula. The test results lie within the curves

$$Q_a = \int_1^2 c_v dT \quad \text{and} \quad Q_a = c_v(T_1 - T_2) - \frac{p_2}{427}(v_2 - v_1).$$

Both sets of curves in figs. 1 and 2 draw nearer as the gas charge increases. While, however, in fig. 2 the efficiency according to equation (5) increases as the charge increases, it decreases according to $\int c_v dT$ as the charge increases, but not so rapidly as it does according to $\int c_v dT - Q_w$. The machine has the ever-recurring characteristic of operating less satisfactorily when too much gas is pumped in. The efficiency falls appreciably with an increasing charge of gas. The actually observed conditions can therefore not be explained by analytical examination according to equation (5). The heat loss and the variation in specific heat must be taken into consideration if the investigator wishes to explain the actual results obtained, and he must picture the process in order to arrive at the basis of these results. By this means science has given us Stodola's gas entropy diagram and Stefan-Boltzmann's equation, which latter has long been known, and confirmed on all sides by recent tests. By purely mathematical investigation and derivation, complicated processes cannot be completely elucidated—nature is too complex for them to be explained by a simple formula.

It will be of some interest to analyse the expansion process, speaking mathematically, not only considering the integral between the top and bottom temperature limits, but also the intermediate values, as different turbine efficiencies are obtained with a variation of the quantity of heat available and a variable heat drop.

Equation 4 of Mangold may be utilised, this being applicable only for constant specific heat, and with the assumption that the heat loss is zero, and also the gas entropy diagram, in which there is no heat loss.

In Table V. the individual values of ΔQ_a, etc., are calculated for stages of 20 per cent., first with $g = 50$ gr. and $t_0 = 150°$ C., and secondly with $g = 90$ gr. and $t_0 = 300°$ C.

In Table VI. the corresponding values are calculated with the help of the gas entropy table. The main results are shown in the curves on fig. 4 for the case $g = 50$ gr. and $t_0 = 150°$ C., and in the curves in fig. 5 when $g = 90$ gr., $t_0 = 300°$ C.

Fig. 5 shows that, when the temperature T_2 is reached, 193·5 calories are rendered available according to Mangold's equation, and 200 calories according to the entropy table, which is practically the same.

But there are considerable differences in the intermediate values. According to equation (4), after the temperature drop is 20, 40, 60, 80 per cent. of the total, Q_a is greater than it should be, according to the entropy table, by 93, 52, 40, 16 per cent.

From this standpoint the momentary efficiency of the turbine has a very important influence on the combined results. When, as is the case for the conditions of equation (4), about 82 per cent. of the total energy is rendered available after the first half of the temperature drop, it is evident that, with

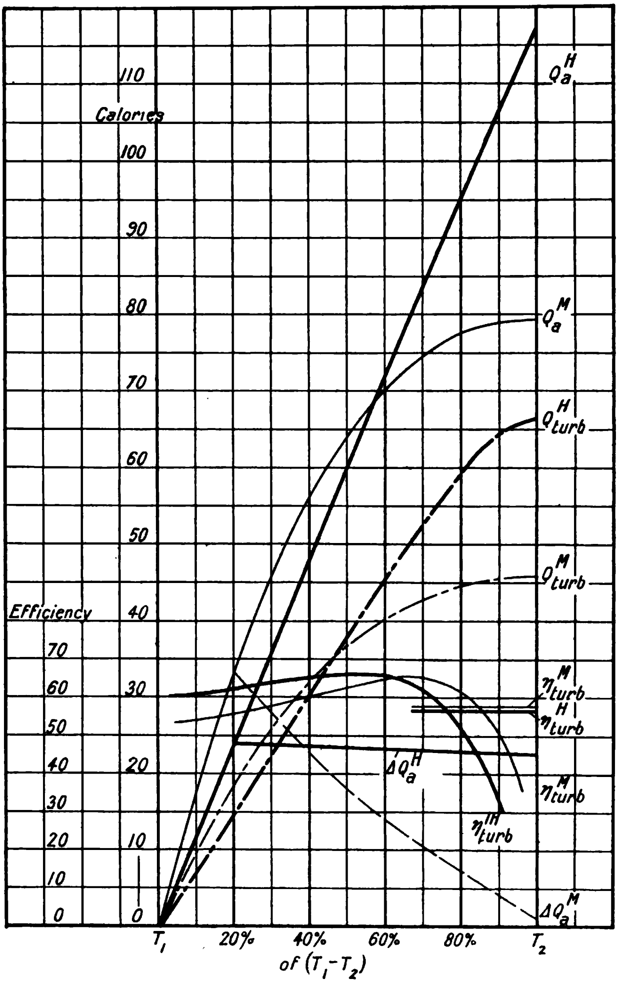


FIG. 4.—$g = 50g$, $t_0 = 150°$ C.

Diagram for intermediate values for $Q^M{}_a = c_v(T_1 - T_x) - c_v(kT_2 - T_x)\left[1 - \left(\frac{T_x}{T_1}\right)^{\frac{1}{k-1}}\right]$

and for $Q^H{}_a = \int c_v dT$ (from entropy diagram).

these assumptions, not much more can be added or subtracted; the turbine efficiency during the second half of the temperature drop is almost negligible compared with that in the first half.

According to the nozzle and blade ratios, more actual energy can be obtained (other conditions being the same) if following equation (4), than according to the entropy table, although both lead to approximately the same result for $Q_a\Big/\frac{1}{2}$.

The momentary turbine efficiency $\eta'_{turb.}$ is derived from a turbine efficiency curve with about 65 per cent. as a maximum for a two-stage wheel, which is usually reached, if not exceeded, in steam turbines. According to Josse and

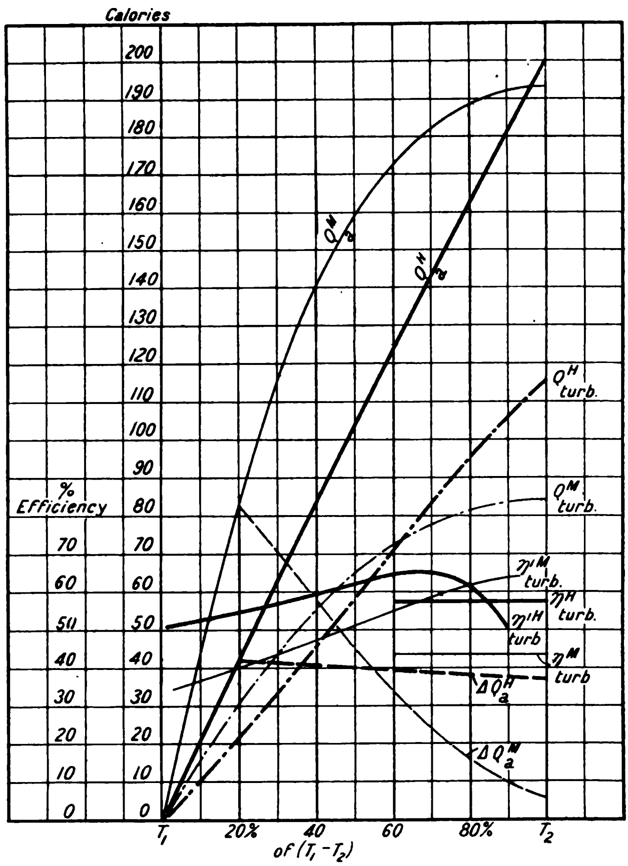


FIG. 5.—$g = 90g$, $t_0 = 300°$ C.

Diagram for intermediate values for $Q^M_a = c_v(T_1 - T_x) - c_v(kT_2 - T_x)\left[1 - \left(\frac{T_x}{T_1}\right)^{\frac{1}{k-1}}\right]$ and for $Q^H_a = \int c_v dT$ (from entropy diagram).

Dr Christlein, it seems doubtful whether, for gas with a speed of propagation of 700 to 800 metres per second, very much better results could not be obtained with a two-stage wheel than with steam having a speed of propagation of about 440 metres per second. Dr Christlein has shown that the efficiency curve of the moving wheel has a definite maximum for the speed of propagation. In a steam turbine, the first stage of the moving wheel works in general on the falling branch of the curve to the right (zero being on the left), whilst for gas, under otherwise similar conditions, the most important stage, from the point of view of energy production, works at the top of the curve.

It cannot, therefore, be doubted that, by a proper choice of the turbine ratios, better efficiencies can be obtained with gas for a two-stage wheel than with steam. Naturally, this can only be shown by special tests.

The average values of $\eta'_{turb.}$ from the examples correspond fairly well with those in fig. 43.

Fig. 5 shows other relationships for $g=50$ gr. and $t_0=150°$ C. It is clearly seen that the differences between the values of Q_a calculated in different ways for the total temperature drop, are partially compensated by the rapidly falling portion of the curve $\eta'_{turb.}$

In any case, these analytical examinations of the expansion process have proved that far too many varying factors influence the final results for a simple mathematical formula to be able to express them.

The question of the control of the temperature fortunately involved no insurmountable difficulties in our gas turbine, as the exhaust temperature never exceeded 450° C.

Summary.

The test results of the gas turbine were compared with the calculated values based on the method previously used by the author, and also with the values obtained by the employment of Stodola's general formula.

These comparisons showed that the test results agreed well with the final theoretical determination of the author, and that therefore Stodola's general formula must be incorrect. Moreover, a purely analytical examination, according to Dr Mangold of Augsburg, also proved the inaccuracy of Stodola's final formula; but, with certain limiting assumptions, Dr Mangold arrived at the same penultimate formula as Stodola. The results of these calculations are drawn in the curves, and lie considerably higher than they should according to Stodola's final formula.

By means of two examples, the expansion process was analysed, using the analytical and graphical method in combination with the entropy diagram, and it was thus clearly shown that the combined conditions are too variable and complicated to be represented in a single mathematical formula. In these examples the average turbine efficiency was deduced from the momentary efficiencies.

It was shown again that only Stodola's graphical method, as used by the author in conjunction with the entropy diagram, allowed the heat process to be followed through, taking into consideration the heat loss and the variation of specific heat. The analytical methods prove quite fully that this is the case.

(co	70g.	(=1680 cb.m.		90g.				(=2160 cb.m.
Init	150°	200°	300°	15°	100°	150°	200°	300°
	423°	473°	573°	288°	373°	423°	473°	573°
Init	1·65	1·70	1·80	1·55	1·70	1·75	1·80	1·90
Vol (1	1·5	66·6	76·3	57·1	67·6	74·2	81·0	92·8
Vol	8·5	133·4	123·7	142·9	132·4	125·8	119·0	107·2
We	4·5	164·0	132·5	262·5	206·0	178·0	155·0	121·0
Mi	: 2·6	1 : $2{\cdot}3_5$	1 : 1·9	1 : 2·9	1 : 2·3	1 : 2·0	1 : 1·7	1 : 1·3
We	4·5	234·0	202·5	352·5	296·0	268·0	245·0	211·0
He	2·6	82·6	82·6	106·2	106·2	106·2	106·2	106·2
He	25·0	352·0	408·0	301·0	359·0	396·0	483·0	504·0
	·.. 61	0·.. 63	0·.. 66	0·.. 59_5	0·.. 63	0·.. 65_5	0·.. 68	0·.. 72
	7·0	88·0	111·0	49·9	67·1	78·0	89·0	113·1
	2·0	440·0	519·0	350·9	426·1	474·0	522·0	617·1
√	0·281	0·292	0·314	0·267	0·289	0·304	0·318	0·345
	2050	2160	2395	1870	2110	2260	2380	2620
Co	94·0	93·5	93·0	94·2	93·6	93·0	92·5	92·0
	7·50	7·25	7·00	9·48	9·00	8·70	8·40	8·00
	1110	1130	1175	1030	1120	1150	1170	1220
	500	750	1500	130	340	500	750	1500
	1,300	56,450	77,800	36,970	55,410	67,100	76,850	100,500
	·.. 630	0·.. 624	0·.. 612	0·.. 703	0·.. 683	0·.. 670	0·.. 665	0·.. 650
Q_r	32·3	35·2	47·6	26·0	37·9	45·0	51·1	65·4
Q_r	27·0	151·0	235·0	73·8	128·0	168·0	208·0	310·0
Q_r	0,400	12,450	19,500	5950	10,480	13,800	17,100	25,600
Q_a	8460	8740	8150	9800	10,200	10,100	10,500	8400
Q_a	03·5	107·0	98·0	121·4	125·0	123·0	128·0	101·2
Q_a	31·8	30·4	24·0	40·4	34·9	31·0	29·5	20·0
	56·8	57·0	56·2	58·0	58·0	58·0	58·0	56·5
	18·1	17·3	13·5	23·4	20·2	18·0	17·1	11·5
	559	534	417	928	802	715	678	448

Weight of G (corresponding of gas per ho		(=1680 cb.m.)		90g. (=2160 cb.m.)				
Initial temperat		200°	300°	15°	100°	150°	200°	300°
Basis of calculati								
Height of trapez		57·8	58·0	60·8	62·0	62·0	62·5	62·5
Side of trapeziu		205·0 250·0	228·5 276·0	175·5 220·0	198·0 246·0	212·0 262·0	225·5 276·0	252·0 304·0
Q_a . .	0	13,150	14,630	12,050	13,750	14,700	15,650	17,300
Q_a . .		160·6	176·3	149·5	168·4	178·1	190·7	209·0
$Q_a : Q = \eta_{comb.}$		45·6	43·3	49·6	46·9	45·0	44·0	41·4
$\eta_{turb.}$		57·2	56·5	57·8	57·0	56·2	55·0	53·0
η_{total}		26·1	24·5	28·7	26·7	25·3	24·2	22·0
		805	756	1140	1060	1010	960	873
Basis of calculati	=0).							
W_2 cub. m./kg.		150·0	175·0	118·0	140·0	155·0	165·0	185·0
W_1 ,, ,,		25·0	28·0	17·0	19·5	22·0	24·0	27·5
$W_2 - W_1$ cub. m		125·0	147·0	101·0	120·5	133·0	141·0	157·5
$v_1 - v_2$ cub. m./k	0	4·46	5·25	3·61	4·30	4·75	5·03	5·62
$\frac{9000}{427}(v_2 - v_1)$		94·0	110·5	76·2	90·8	100·0	106·1	118·5
Q_a . .		66·6	65·8	73·3	77·6	78·1	84·6	90·5
$Q_a : Q = \eta_{comb.}$		18·9	16·1	24·3	21·6	19·7	19·5	17·9
$\eta_{turb.}$		51·7	51·5	53·0	54·0	54·0	54·9	56·2
η_{total}		9·8	8·3	12·9	11·7	10·6	10·7	10·0
		303	256	512	464	421	425	397

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